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Technical Memorandum

HEAT PIPE APPLICATION FOR SPACECRAFT THERMAL CONTROL

by D. K. ANAND and R. B. HESTER

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THE JOHNS HOPKINS UNIVERSITY • APPLIED PHYSICS LABORATORY

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HEAT PIPE APPLICATION FOR SPACECRAFT THERMAL CONTROL

by D. K. ANAND and R. B. HESTER

THE JOHNS HOPKINS UNIVERSITY • APPLIED PHYSICS LABORATORY
8621 Georgia Avenue, Silver Spring, Maryland 20910
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The Johns Hopkins University
APPLIED PHYSICS LABORATORY
Silver Spring, Maryland

ABSTRACT

A heat pipe is a device which exhibits an extremely high effective thermal conductivity by means of two-phase fluid flow with capillary circulation.

The primary objective of the experimental program was to determine a suitable method of control for the heat pipe and to establish suitable wick/fluid configurations for the various temperature ranges of interest.

The primary objective of the prototype program was to provide design, construction, testing for verification, and flight hardware specifications of a heat pipe applicable to thermal control of a spacecraft or a spacecraft subsystem. Thus, a thermal design improvement for spacecraft could be proposed; in addition, thermal resistances of heat pipes could be derived.

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I. Introduction

A heat pipe is a device which exhibits an extremely high effective thermal conductivity by means of two-phase fluid flow with capillary circulation. In simplest form the device consists of a length of tubing, sealed at both ends, containing an annular wick and a small amount of working fluid. Heat enters one end of the pipe (the evaporator section) and evaporates a portion of the fluid. The vapor then flows to the opposite end of the pipe (the condenser section) where it is condensed. The condensate is returned to the evaporator by the capillary pumping action of the wick.

This study and development work on heat pipe application for spacecraft thermal control was sponsored by the National Aeronautics and Space Administration-Office of Advanced Research and Technology in accordance with APL/JHU letter TS-1343 dated 19 September 1966. This letter was approved by NASA letter RNW dated 10 October 1966 addressed to the Laboratory.

II. Objectives

The primary objective of the experimental program was to determine a suitable method of control for the heat pipe and to establish suitable wick/fluid configurations for the various temperature ranges of interest.

The primary objective of the prototype program was to provide design, construction, testing for verification, and flight hardware specifications of a heat pipe applicable to thermal control of a spacecraft or a spacecraft subsystem. Thus a thermal design improvement for spacecraft could be proposed. In addition thermal resistances of heat pipes could be derived.

III. Description of Tests

A. Experimental Design

To accomplish the objectives, two heat pipes were fabricated and were subsequently instrumented. The heat pipes are of stainless steel construction and are provided with valving to facilitate vacuum pumping and fluid charging. Controllable resistance heating was utilized to bring heat into the system while heat removal from the condenser will be accomplished by means of a constant temperature bath. Appropriate temperature and pressure instrumentation is provided to measure heat fluxes and overall heat transfer coefficients. Insulation is provided over the evaporator and isothermal sections of the heat pipe to minimize heat losses to the atmosphere.

1. Control Techniques

Various control techniques were investigated. These can be categorized as follows:

a. Regulation of vapor flow.
b. Control of condenser parameters.
c. Introduction of non-condensable gases.
d. Control of the pressure of the working fluid.
e. Regulation of fluid flow through the wick.

Only the first three methods were investigated in detail. Earlier work at the Applied Physics Laboratory has proved that heat pipe operation can be effectively stopped by providing a complete wick discontinuity (item e); however, this method may not be the most suitable for space applications. Refer to appendices.

2. Experiments

The following experiments were conducted:

2. Experiments (cont'd.)

a. Vapor regulation: The flow area will be progressively blocked near the condenser while holding other variables constant. Curves will be obtained showing the variation of average heat pipe temperature with percent open area for given heat inputs and constant condenser parameters.

b. Variation of condenser parameters: Condenser area and temperature will be varied while holding other parameters constant. A family of curves showing the effects of each parameter on system performance will be obtained.

c. Injection of a non-condensable gas: The heat pipe will be operated with measured amounts of a non-condensable gas injected to determine the effect on temperature.

d. Wick/fluid configurations: Different wicking materials, such as stainless steel screen and porous stainless steel, will be utilized with such working fluids as ethyl alcohol, distilled water and freon. Operation will be at high, as well as low, temperatures to verify whether the performance is satisfactory.

B. Prototype Design

Figure 1 is a schematic diagram of the prototype arrangement. As shown, the major items are the refrigeration unit, the cooling tank, and the heat pipe assembly. An absolute pressure transducer, in conjunction with a wheatstone bridge and voltmeter, is used to measure vapor pressure. Copper-constantan thermocouples sense temperatures on the outside of the heat pipe and at various locations within the insulation. These temperatures were automatically recorded by a 24-point recorder.

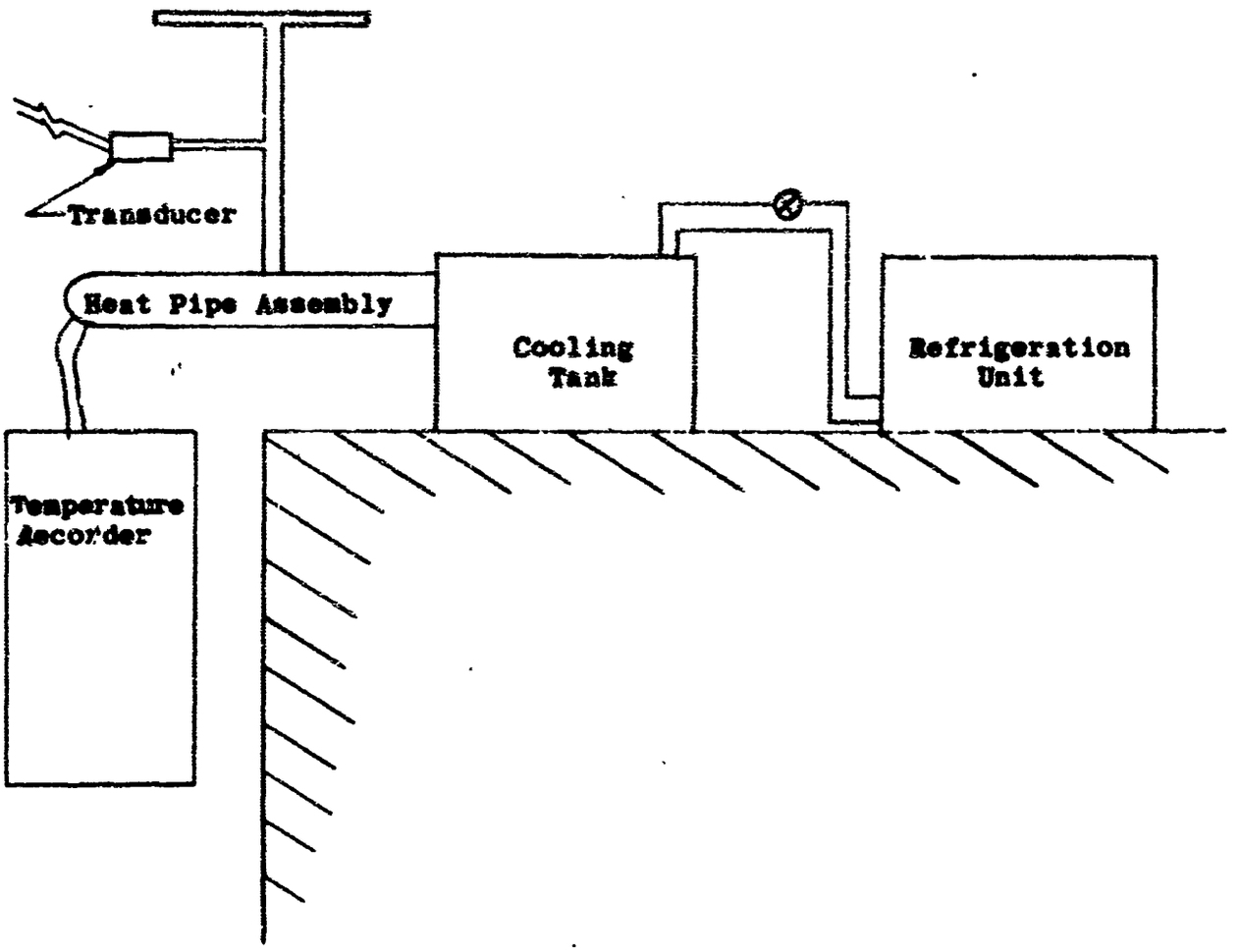


Fig. 1 EXPERIMENTAL ARRANGEMENT

The refrigeration unit is a constant flow device equipped with two evaporators. The main evaporator is located in the cooling tank and serves to remove heat from the fluid therein. An auxiliary evaporator is located within the cabinet of the refrigeration unit. Temperature control is accomplished by means of a sensing bulb and a bellows assembly. When the temperature of the liquid reaches the control setting, a solenoid valve is energized, diverting the refrigerant from the main to the auxiliary evaporator.

The cooling tank is filled with a mixture of water and antifreeze. A specially-designed split seal fastens to the condenser end of the heat pipe and to the wall of the tank to maintain water-tight integrity. The tank is insulated on all sides to reduce heat leakage.

At the evaporator end of the heat pipe assembly, a resistive heating coil, wound on a hollow aluminum cylinder, is used to supply heat to the pipe. Power to the coil is furnished by a DC power supply which has suitable controls to vary the power as desired.

IV. Test Procedures

A. Experimental Design

(Refer to appendices)

B. Prototype Design

Calibration of the absolute pressure transducer was accomplished prior to filling the heat pipe. Briefly, the procedure was to evacuate the pipe with a vacuum pump and balance the wheatstone bridge. Next, the vacuum pump was removed and the pressure was allowed to equalize with the atmosphere. The voltage and atmosphere pressure were recorded and a linear relationship was assumed to hold. This correlation was checked at several other pressures during the normal course of atmosphere pressure variation.

After filling and sealing, the heat pipe was instrumented, insulated, and the condenser end inserted into and sealed to the cooling tank. The refrigeration unit was energized and set to a prescribed level. The heat input was then adjusted by means of the power supply and the electrical input measured. Pressure and temperatures were recorded until steady state conditions were attained. The heat input level was then changed while holding all other parameters constant. To verify that the data were repeatable, certain runs were conducted again after an interval of a few days.

V. Test Results

A. Experimental Design

(Refer to Appendices)

B. Prototype Design

To correlate the data, an overall heat transfer coefficient, U, is defined by:

$$q = UA\Delta T$$

where ΔT = the difference between the mean wall temperature of the evaporator or condenser and the fluid

A = the heat transfer area

U is thus seen to include conduction as well as convective effects. A dimensionless heat transfer coefficient, $\frac{UL}{K}$, is then calculated and correlated with a dimensionless heat flow, $\frac{q}{q_o}$, where:

$$q_o = \frac{S\Delta T\mu D_e}{Pr}$$

and S = entropy

μ = absolute viscosity

D_e = equivalent diameter

Pr = Prandtl number

Figures 2 and 3 show this correlation for the evaporator and condenser of the heat pipe tested. The curves are typical of convective heat transfer data and substantiate the choice of correlation.

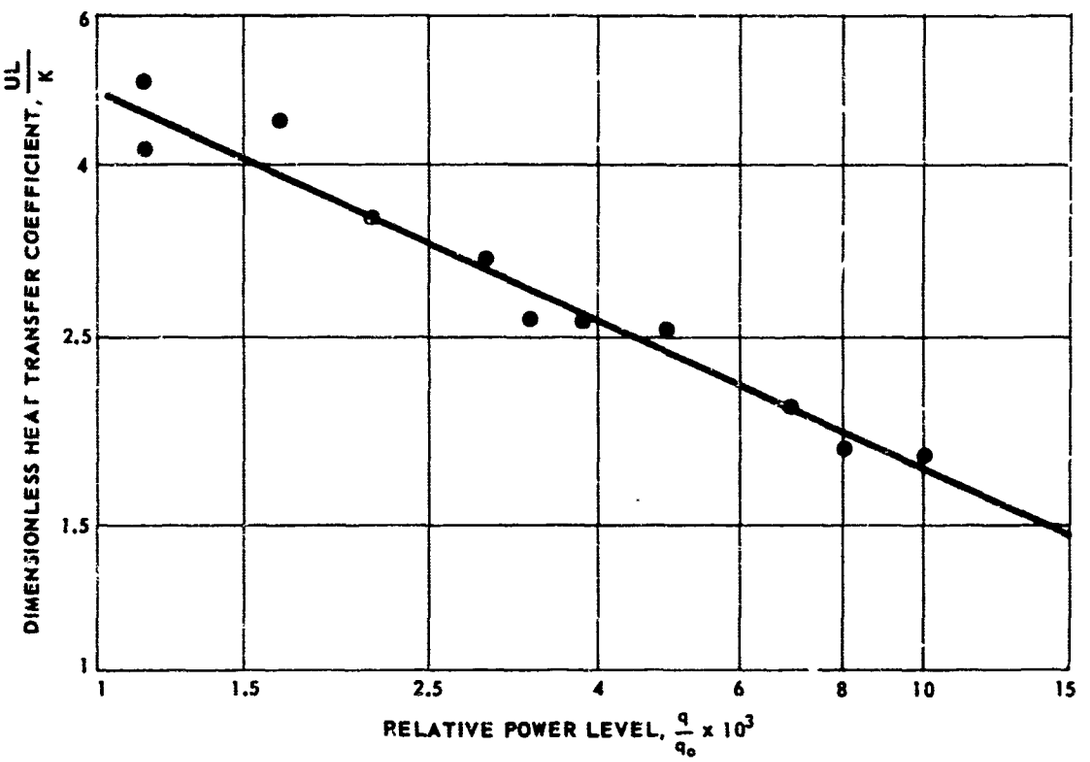


Fig. 2 VARIATION OF EVAPORATOR HEAT TRANSFER COEFFICIENT WITH POWER LEVEL

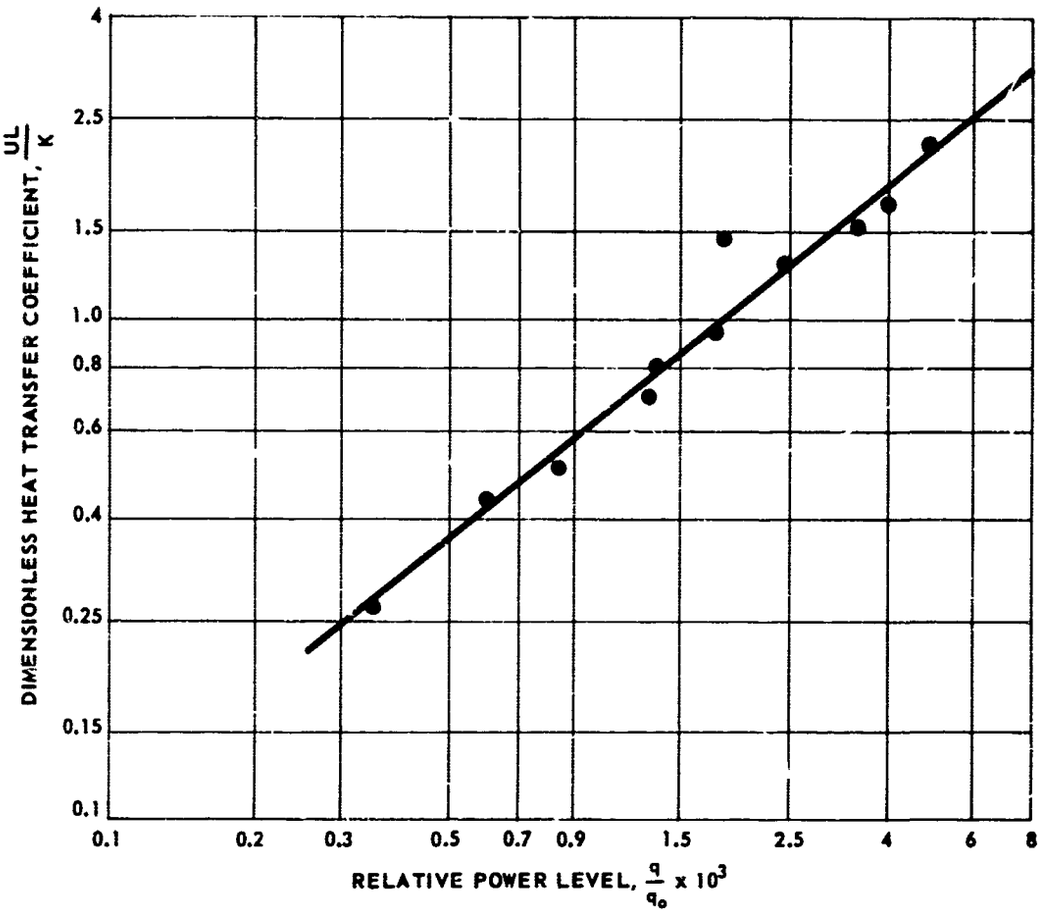


Fig. 3 VARIATION OF CONDENSER HEAT TRANSFER COEFFICIENT WITH POWER LEVEL

To calculate the thermal resistance at a given heat flux, consider the analogy between the expression defining U and Ohm's law:

$$q = UA\Delta T$$

$$I = \frac{E}{R}$$

The emf, E, corresponds to ΔT and the thermal resistance is thus $R = \frac{1}{UA}$. The thermal resistance of the isothermal section of the heat pipe is negligible. Hence, the total resistance of the heat pipe consists of the series resistances of the evaporator and condenser:

$$R_p = R_e + R_c = \left|\frac{1}{UA}\right|_e + \left|\frac{1}{UA}\right|_c$$

For comparison, the thermal resistances of lengths of pure copper bars of weight equal to that of the heat pipes were calculated. The resistance of a thermal conductor is given by:

$$R = \frac{L}{KA}$$

where L is the length and K is the thermal conductivity.

At a heat flow of 100 BTU/hr., the relative resistances of the heat pipes and copper bars are:

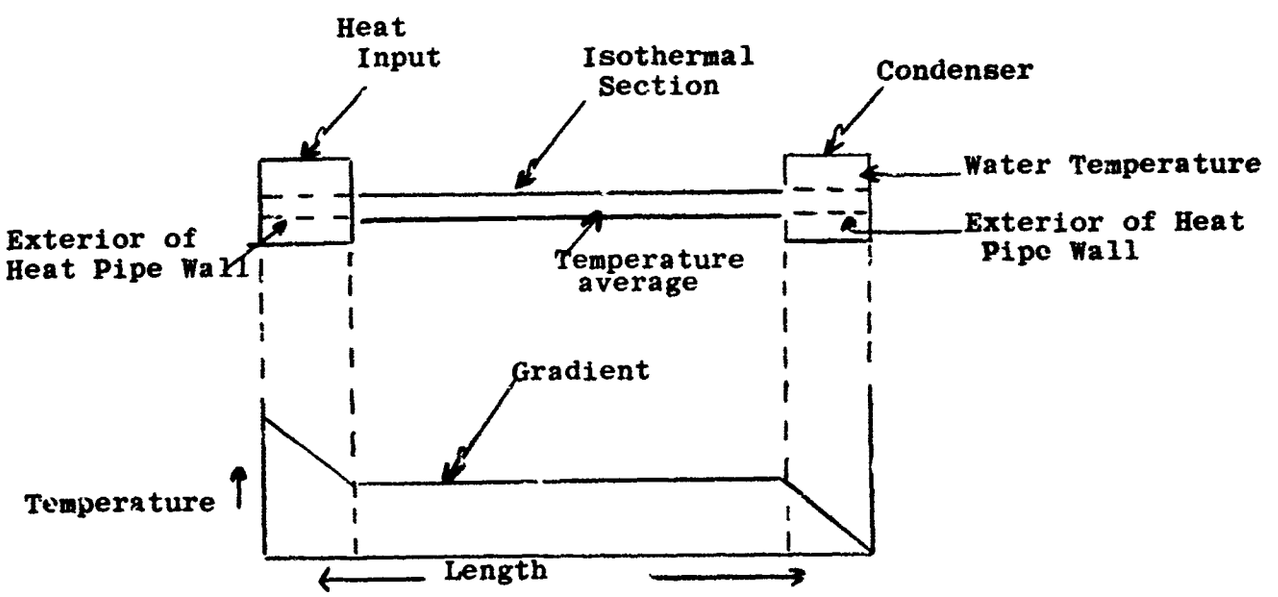
$$\frac{R_{c_u}}{R_p} = 27.6 \quad \text{for an 18" heat pipe}$$

$$\frac{R_{c_u}}{R_p} = 58.2 \quad \text{for a 38" heat pipe}$$

To summarize, Figures 2 and 3 present data correlations in dimensionless form. Using this data, the thermal resistances of the two proposed GEOS-B heat pipes were calculated and compared to the resistances of lengths of pure copper of equal weight. For a power level of 100 BTU/hr., the thermal resistances of the copper bars were about 28 and 58 times that of the short (18") and long (38") heat pipes, respectively.

Figure 4 below indicates the prototype bench test setup and relation of temperature measurements vs pipe length. The prototype heat pipe performance is shown in Figure 5. Varying power heat inputs are shown on each curve.

Fig. 4 PROTOTYPE BENCH TEST SETUP AND RELATION OF TEMPERATURE MEASUREMENTS VS PIPE LENGTH



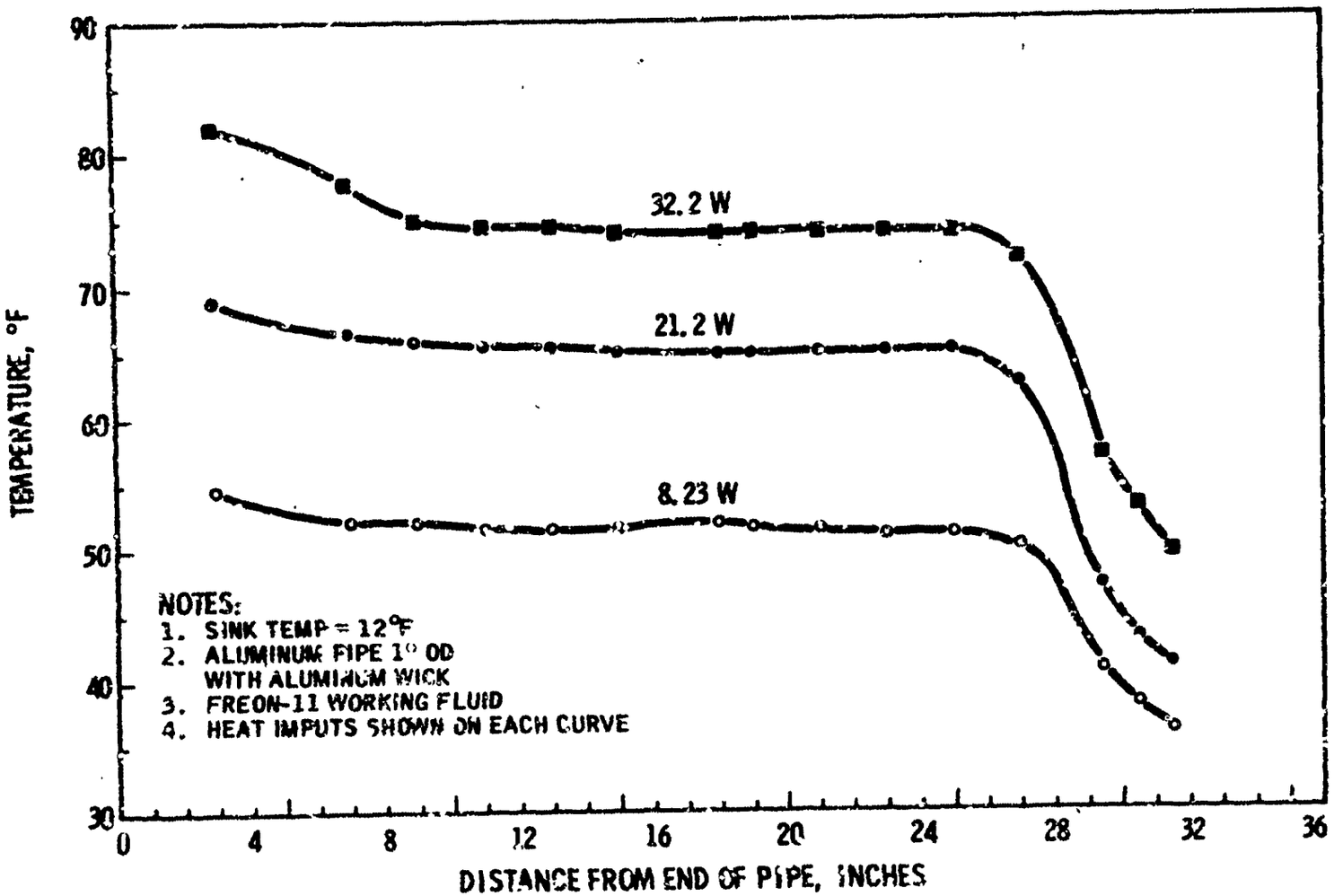


Fig. 5 PROTOTYPE HEAT PIPE PERFORMANCE

VI. Flight Hardware

A. Flight Hardware Performance Specification

1.0 Scope

This specification delineates the performance requirements, inspection and test procedures and qualification for flight of heat pipes for the spacecraft.

2.0 Applicability

Provisions of this specification shall apply to prototype and flight units in both the "instrumented" and "non-instrumented" configurations. In the "instrumented" configuration the heat pipes are equipped with absolute pressure transducers. In the "non-instrumented" configuration, the pressure transducers are removed, the center tube crimped and welded and the protective cap installed and welded to the boss. (See Figures 6 and 7.)

3.0 General Requirements

3.1 Initial evacuation, filling and sealing of the heat pipes shall be accomplished with a minimum of air leakage into the pipes. The maximum permissible air volume for the heat pipes shall be that volume which results in a partial pressure due to air of 2 mmHg per inch of length at a temperature of 70°F.

3.2 There shall be no additional leakage of air into the heat pipes while in the instrumented configuration.

3.3 The heat pipes shall be designed to operate satisfactorily throughout the useful life of the satellite. A failure of one or both heat pipes,

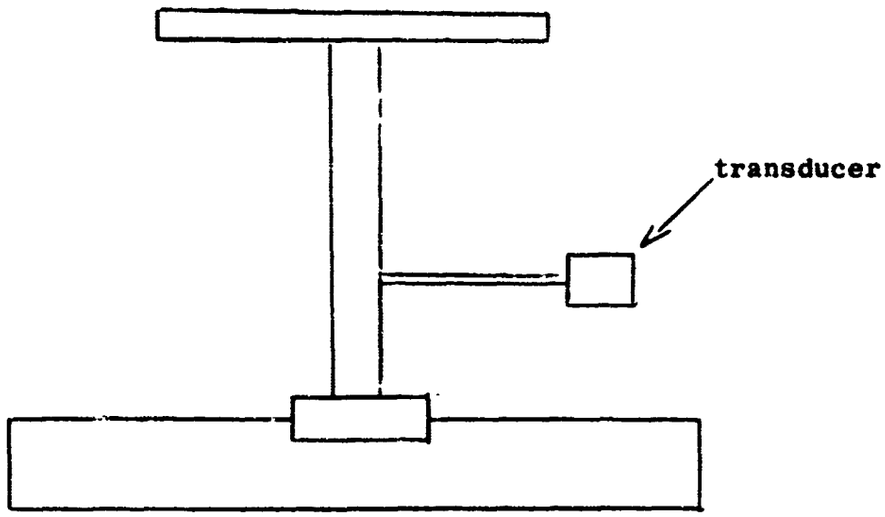


Fig. 6 INSTRUMENTED CONFIGURATION

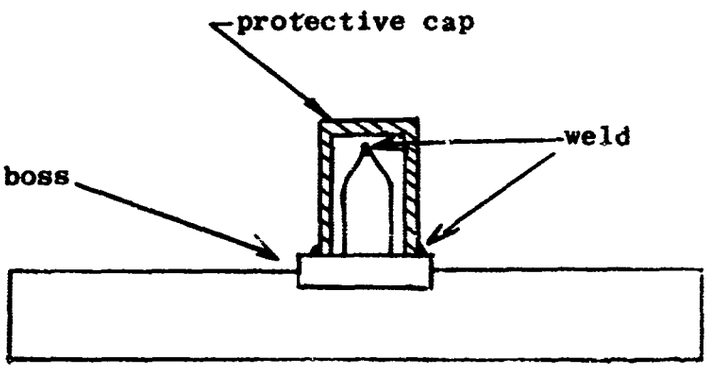


Fig. 7 NONINSTRUMENTED CONFIGURATION

3.0 General Requirements (Cont'd)

3.3 however, shall not be deleterious to satellite performance.

4.0 Performance

4.1 The heat pipes shall exhibit an effective thermal conductivity of at least 2,200 BTU/hr ft °F over an operating range of from 0°F to 130°F. The effective thermal conductivity shall be calculated using temperatures measured on the exterior of the pipe. Temperatures measured at locations closer than two inches from the end of the evaporator section or from the beginning of the condenser section shall not be used for this calculation.

4.2 The heat pipes shall transmit up to 30 watts of heat over a temperature range of from 0°F to 130°F at the specified effective thermal conductivity.

5.0 Inspection and Test

5.1 Prior to evacuating, filling and sealing, the heat pipes shall be subjected to pressure and leak tests. The heat pipes shall withstand a pressure of 125 psig of dry nitrogen gas without leakage for a period of 30 minutes. Following the pressure test, the pipes shall be subject to a helium leak test. There shall be no detectable leakage.

5.2 Bench testing of the pipes in the instrumented configuration shall be accomplished after the pressure and leak tests. The pipes shall meet all requirements specified in Paragraphs 4.1 and 4.2.

5.0 Inspection and Test (Cont'd)

5.3 Bench testing shall be accomplished by utilizing equal areas for the condenser and evaporator. Resistance heating shall be used for the evaporation of fluid while a controlled temperature bath will be used as a heat sink. The heat pipe shall be appropriately insulated. Instrumentation shall be provided to measure the temperature distribution along the outside of the heat pipe as well as losses to the atmosphere. For each run the power level and the heat sink temperature shall be maintained constant. When steady state conditions are attained, the experimental parameters may be changed and a new run began.

6.0 Qualification for Flight

6.1 After the completion of the tests specified in Paragraphs 5.1 and 5.2, final crimping and sealing shall be accomplished. The effective thermal conductivity shall again be measured as per Paragraph 4.1 at power levels up to 30 watts. A degradation of effective thermal conductivity of as much as 5% of the original value is permitted.

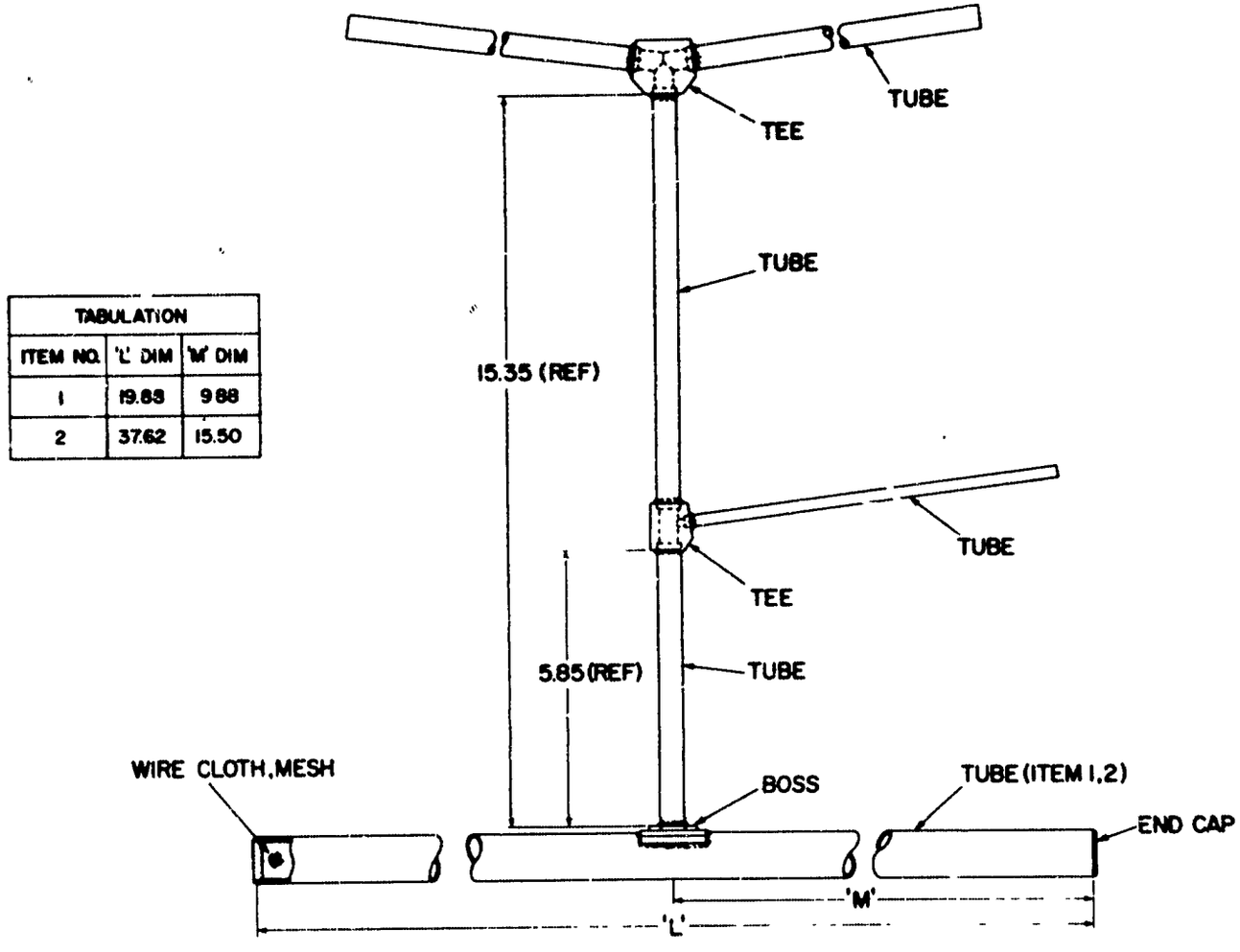
6.2 Vibration tests shall be conducted on the heat pipes at the frequencies and amplitudes specified in the Design Data Sheets. Following the completion of the vibration tests the effective thermal conductivity shall again be measured. A degradation of 1/2% of the value as measured during the tests of Paragraph 6.1 is permitted.

6.3 Thermal vacuum testing shall be accomplished with the heat pipes installed in the satellite in accordance with the applicable Design Data Sheets.

VI. Flight Hardware (Cont'd)

B. Flight Hardware Design Example

Figure 8 presents an example of a spacecraft heat pipe design assembly while Figure 9 shows a typical layout for electronic package connection.



TABULATION		
ITEM NO.	'L' DIM	'M' DIM
1	19.83	9.88
2	37.62	15.50

Fig. 8 HEAT PIPE ASSEMBLY

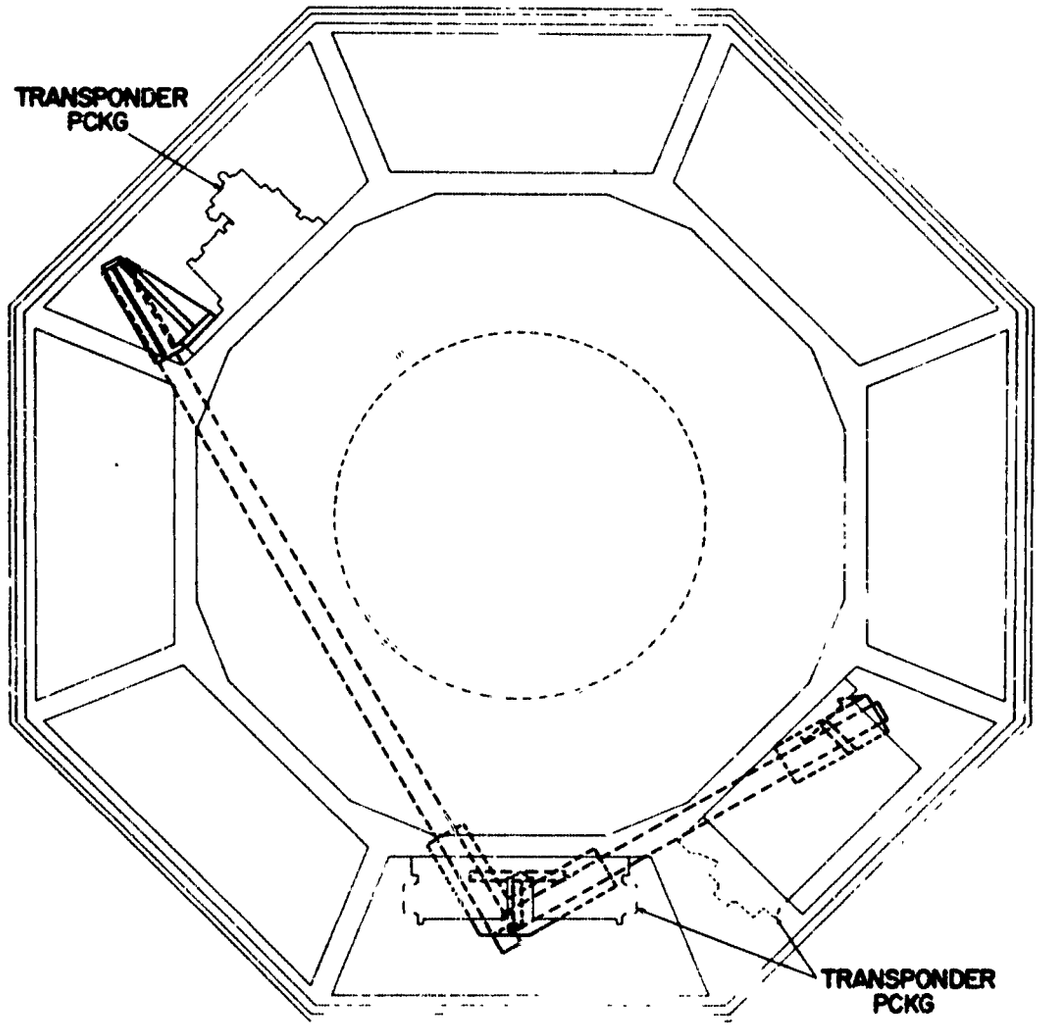


Fig. 9 TYPICAL HEAT PIPE LAYOUT FOR SPACECRAFT THERMAL CONTROL

VII. Summary and Conclusions

The test results for both the early experimental work and prototype effort show excellent correlation as to performance and control of a heat pipe design. The data also clearly demonstrates its proposed use to provide thermal control in a spacecraft application.

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ACKNOWLEDGMENT

Acknowledgment is made to R. Harkness and R. E. Fischell (APL) for development effort of the prototype heat pipe; R. Willison, G. C. Weiffenbach, C. Wingate, S Willis, A. Reymann (APL) for their analysis and design reviews of spacecraft applications; and to T. Wyatt (APL) for suggestions and comments throughout the subject effort.

Appendix A

EFFECTS OF CONDENSER PARAMETERS ON HEAT PIPE OPTIMIZATION

by

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Appendix A

EFFECTS OF CONDENSER PARAMETERS ON HEAT PIPE OPTIMIZATION

In most applications of heat pipes the maximum heat transport is dependent upon liquid circulation due to capillary forces in the wick.[1] For a specific capillary structure the local pressure difference must be

$$p_v(z) - p_\ell(z) \leq \frac{2\,\sigma \cos\theta}{r_c} \tag{1}$$

where the equality would yield maximum heat transport. For low Reynolds number Re, substituting the appropriate pressure drops,[1,2] Eq. (1) becomes

$$\frac{4\,\mu_v(\ell_e + L)Q}{\pi\,\rho_v\,r_v^4\,\lambda} + \frac{\omega\,\mu_\ell(L + \ell_e)\,Q}{2\pi(r_w^2 - r_v^2)\rho_\ell\,e r_c^2\,\lambda} - \frac{2\,\sigma\cos\theta}{r_c} = 0, \tag{2}$$

for maximum heat transport. Here r_v is equal to the inner radius of the heat pipe (r_w) minus the thickness of the wick; i.e., it is the inner radius of the wick. Optimum values for r_c and r_v can be obtained from Eq. (2), all other properties being fixed. The value of r_c that would maximize Q is determined from $\partial Q/\partial r_c = 0$, where Q is defined by Eq. (2). This value when resubstituted into Eq. (2) yields

$$\frac{2\,\mu_v\,\mu_\ell(\ell_e + L)^2\,\omega\,Q^2}{\rho_\ell\,\rho_v\,\pi^2\,e\,\lambda^2\,\sigma\,r_v^4(r_w^2 - r_v^2)\cos\theta} - \sigma\cos\theta = 0 \tag{3}$$

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It is clear that the maximizing of Q now requires the evaluation of $\partial Q/\partial r_v = 0$. This constraint yields $(r_v/r_w)^2 = 2/3$ which when substituted into Eq. (3) will yield

$$Q = \frac{2\pi r_w^3 \lambda \sigma \cos\theta}{3(L + \ell_e)} \left[\frac{e\,\rho_v\,\rho_\ell}{6\omega\,\mu_v\,\mu_\ell}\right]^{1/2} \qquad (4)$$

Under steady state conditions, the heat Q transferred through the condenser may be denoted by $[k\,\pi\,D(T_v - T_o)/\ln\frac{r_o}{r_v}]\,L$ where K,D,ΔT,L are the effective conductivity, heat pipe diameter, radial temperature drop from inside to outside surface, and the condenser length respectively. This Q may also be written as CL where C becomes $(k\,\pi\,D\,\Delta T/\ln\frac{r_o}{r_i})$. Substituting L = Q/C into Eq. (4) leads to a quadratic in Q, which yields the solution

$$Q_{optimum} = [\tfrac{1}{3}\,C\,\pi\,r_w^3\,\lambda\,\sigma\cos\theta(\tfrac{2}{3}\,\frac{\rho_v\,\rho_\ell}{\mu_v\,\mu_\ell}\,\frac{e}{\omega})^{1/2} + \tfrac{1}{4}\,C^2\,\ell_e^2]^{1/2} - C\,\ell_e/2 \qquad (5)$$

From Eq. (5), it is seen that the operation of the heat pipe, is constrained chiefly by condenser parameters. Extensive experimental data obtained at the Applied Physics Laboratory bears this out. Figure 1 shows the "heat pipe regime"* temperatures as a function of the condenser parameter C. The experimental data is for the following heat pipe.

*That segment of the heat pipe over which the temperature drop of the vapor is small, of the order of 1-2°F. This segment consequently exhibits a very high thermal conductivity.

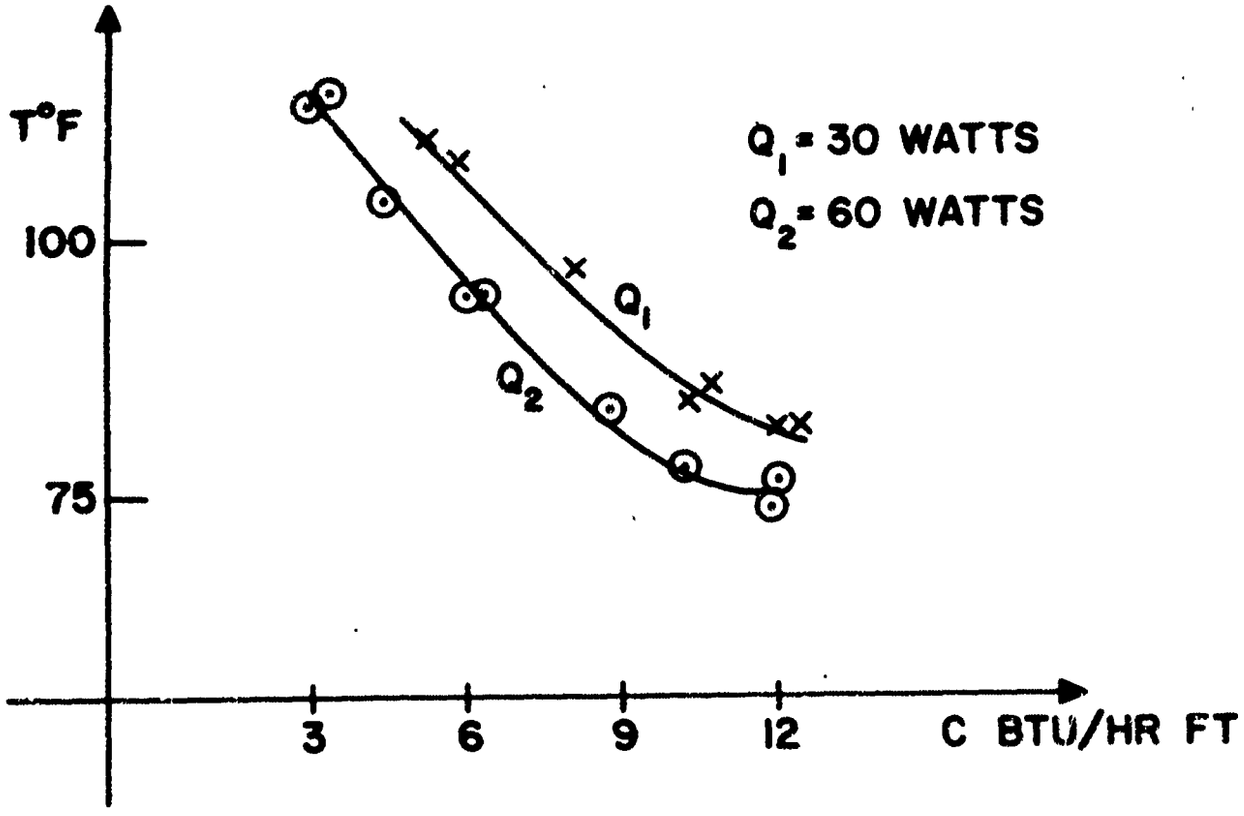

Figure 1: EFFECT OF CONDENSER PARAMETERS ON HEAT PIPE REGIME TEMPERATURE. CONDENSER SURFACE TEMPERATURE = 54° F HEAT PIPE REGIME TEMPERATURES EXHIBIT A MAXIMUM DROP OF 2° F OVER 22 INCHES.

Appendix A

$$(r_w - r_v) = 0.063 \text{ inches}$$

$$\epsilon = 0.68$$

$$r_v/r_w = 0.815$$

$$r_c = 3.25 \times 10^{-3} \text{ inches}$$

$$Q = 30 \text{ watts}$$

$$\omega = 15$$

fluid = ethyl alcohol

The optimum heat transport that could be achieved is 725 watts with 0.1 effective conductivity, 60° contact angle, 1.5592×10^{-3} lb_f/ft surface tension and a radial temperature drop of 20° F across condenser surface. The condenser surface area required is 5 square feet. The variation of condenser parameter C may be achieved by flooding, introduction of noncondensible gases, or manually varying surface area. The reported data is achieved by flooding and varying surface area. The effect of noncondensible gases is similar although more dramatic.

The above comments are based on nonradiative type condensers. In applications where the heat is radiated away from the exterior surface of the condenser, an interesting situation may develop. We have given the name "temperature choking" to this effect and offer the following explanation. The equilibrium temperature of the vapor (and consequently the evaporator section) for a given power dissipation is determined by the temperature of the outer surface of the condenser. For radiative condensers the equilibrium equation becomes

$$F\,A\,\sigma(T_o^4 - T_a^4) = L\,k\,\pi\,D(T_v - T_o)/\ln r_o/r_v \qquad (6)$$

Now, since the heat is removed from the outer surface by radiation, there is no longer a strong boundary condition on the temperature of the outer surface as there is in the case of a condenser bath. Consequently, the outer temperature will adjust itself so that in equilibrium the heat radiated equals the heat input. If the heat flow through the condenser is increased, the temperature of the outer condenser surface must now increase to radiate away the extra heat; and since the temperature difference across the composite condenser wall determines the heat conducted through the wall, the temperature of the vapor core must increase even more. Thus, the temperature response of the vapor to heat flow increases should be quite different in this case from the case of a condenser bath, where bath temperatures may be independently varied.

From a design viewpoint, this requires high C values in Eq. (5).

Nomenclature

C = condenser parameter, $k \pi D(T_v - T_o)/\ln r_c/r_v$, BTU/hr ft

D = heat pipe diameter, ft

e = porosity of wick, $(\rho_f - \rho_{w_i})/\rho_f$

F = radiation shape factor

ℓ = length of evaporator, ft

L = length of condenser, ft

Re = Reynolds number, $\frac{Dov}{\mu}$

p = pressure, lb_f/sq. ft^2

Q = heat flow, BTU/hr or watts

r = radius, ft

T = temperature, ^oF

V = velocity, ft/sec

z = axial heat pipe coordinate

ω = constant, property of wick, due to the tortuous path taken by fluid flowing through the pores. Varies between 8-20

θ = contact angle of fluid, radians

λ = latent heat of vaporization, BTU/lb_m

μ = coefficient of viscosity, lb_f sec/ft^2

ρ = density, lb_m/ft^3

σ = surface tension, lb_f/ft

Subscripts

a = ambient

c = wick pore

f = fluid

Subscripts (Cont'd)

ℓ = liquid

v = vapor

w = wall, inside

o = wall, outside

Acknowledgement

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References

1. Cotter, T. P., "Theory of Heat Pipes," LA-3246-MS, Los Alamos Scientific Laboratory, Los Alamos, New Mexico, (1965).

2. Anand, D. K., "On the Performance of a Heat Pipe," Journal of Spacecraft and Rockets, May 1966.

Appendix B

ON THE PERFORMANCE OF A HEAT PIPE

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NOMENCLATURE

A	=	Area of wick, ft^2
C	=	Specific heat, BTU/lb. °F
D	=	Diameter, ft^2
g	=	gravity, ft/sec^2
h	=	boiling heat transfer coefficient for wick = $Q/A(T_v-T_w)$
K	=	thermal conductivity, BTU/ft. hr. °F
L	=	Length, ft.
N_p	=	Pressure number
Pr	=	Prandtl number = $C\mu/K$
p	=	pressure, lb/ft^2
Q	=	heat, BTU/hr.
Re	=	Reynold number = $\rho vD/\mu$
r	=	radius, ft.
St	=	Stanton number = $h/cw = \varphi\ Pr^a Np^b Re^c$
T	=	temperature, °F
v	=	Velocity, ft/sec
w	=	Mass flow rate = $Q/A\epsilon\lambda$, lb/sec.
X	=	Axial direction of heat pipe
φ, π, ω	=	Constants
ϵ	=	Porosity of wick = $(\rho_f - \rho_{wi})/\rho_f$
ρ	=	Density, lb/ft^3
μ	=	Viscosity, lb/hr.ft.
λ	=	Latent heat of vaporization, BTU/lb.
σ	=	Surface tension, lb/ft.

SUBSCRIPTS

f	=	wick fiber
ℓ	=	liquid
p	=	pore in the wick
v	=	vapor
w	=	wall
wi	=	wick

INTRODUCTION

The requirements of cooling in the space environment have led researchers to study various coolants and high thermal conductivity devices. The heat pipe described in this note (Fig. 1) is a self-contained device that exhibits a very high effective thermal conductivity.[1] It consists of a sealed tube with wick material in contact with the internal heat transfer surface. The operation is based on the evaporation of a liquid in the evaporator section and subsequent flow in the core towards a region of low pressure. In the condenser, the liquid is condensed and flows back to the evaporator through the wick by capillary pumping to continue the cycle. Under steady state conditions the pressure in the evaporator section is slightly less than the vapor pressure of the adjacent liquid, thereby assuring continuing evaporation. In the condenser section, the opposite holds, assuring continuing condensation. Owing to this liquid-vapor interface, the radius of curvature of meniscus in evaporator recedes, and that in the condenser increases. In the condenser, especially if there is excess fluid, the radius of meniscus may be infinite

The transient behavior has not been studied extensively, but it would seem that this phase will not present any great difficulty. It is clear, however, that there will be an upper limit to the heat flow through the pipe, because liquid depletion rate in the evaporator must exceed the recirculation rate by capillary pumping.

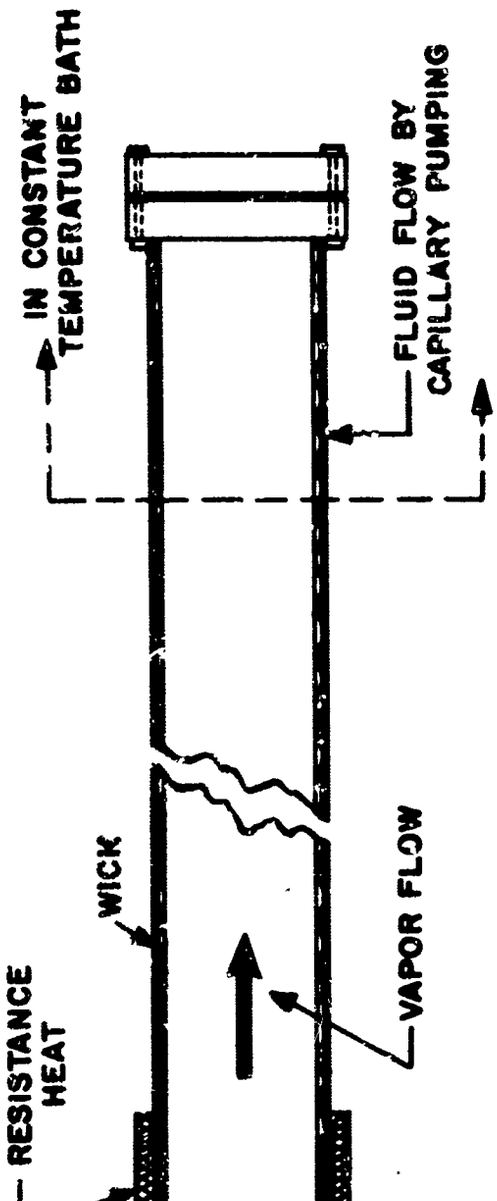

Fig. 1 SEALED HORIZONTAL PIPE FOR WICK BOILING. (INSULATION NOT SHOWN)

DYNAMIC OPERATION

The heat pipe comprises two domains: The vapor core and the fluid annulus (fluid flowing through the wick). The flow conditions in these regions merit separate attention. The vapor flow in the core is similar to flow with injection or suction through a porous wall, since a liquid and vapor are continually changing phase at the interface. Cotter[4] has explained the dynamics of vapor flow and we largely use his explanation here.

Several different vapor flow regimes may be obtained based on the Reynolds number referred to the vapor core diameter. For $Re_v << 1$, the velocity profile is parabolic and flow is similar to Poiseuille flow. The pressure decreases in the direction of flow with a gradient larger than that of Poiseuille flow in evaporation and smaller in condensation. Flow properties are obtained using constant Re_v; for almost all cases of present interest this is justified. For $Re_v << 1$, the velocity profile is no longer parabolic, but vapor pressure still decreases in flow direction, and the properties of flow may still be obtained as above.

For Re_v = constant $<<1$ the pressure is[3]

$$dp/dx = (8\ \mu w_v/\pi \rho r^4)\ (1 + \frac{3}{4} Re_v + \ldots) \qquad (1)$$

Since this drop is small, the first term usually suffices for determining the pressure drop. It is appropriate to remark that the temperature drop may be obtained using the Clausius-Clapeyron equation.

The liquid flow through the annulus is quite different from the core. The momentum equation for incompressible steady flow is:

$$\nabla p = \rho g + \mu \Delta^2 v - \rho v \Delta v \qquad (2)$$

If an average velocity (v) is defined over an area of wick which includes the solid structure, then the velocity within the pore is $(v)/\epsilon$, where ϵ is the wick porosity. Observing that the velocity on the pore surface vanishes and is of the order $(v)/\epsilon$ within the passage, the following approximations may be made:

$$\rho v \nabla v \longrightarrow \rho (v)^2 / r\epsilon^2, \quad \mu \nabla^2 v \longrightarrow \mu (v) / \epsilon r^2 \tag{3}$$

Since (v) is small, equation (2) becomes

$$\nabla p \cong \rho g + \mu \nabla^2 v \tag{4}$$

and the ratio of the two expressions in (3) yield a liquid Reynold number based on (v). Although the last term is neglected in the expression for the pressure drop, this Re_ℓ will be used in the performance correlation; experiment shows that indeed $Re_\ell \ll 1$. From equations (3) and (4) the pressure drop for a horizontal pipe becomes

$$\Delta p_\ell = \omega \mu Q X / 2\pi (r_w^2 - r_v^2)\, \rho \epsilon r_p^2 L \tag{5}$$

where ω is a constant depending upon the capillary structure. The vapor pressure drop has also been computed from equation (1) considering only the first term and is

$$\Delta p_v = -4\mu Q X / \pi \rho r_v^4 L \tag{6}$$

It is assumed that the region between the condensing and evaporator section is perfectly insulated and that $Re_v \ll 1$.

CORRELATION AND EXPERIMENTAL RESULTS

As an extension to the above study, one can correlate the heat transfer coefficients in the evaporator section and thereby obtain a rather good indication of the engineering performance.

Since the problem is essentially that of wick boiling and condensing, in the evaporator and condenser, dimensionless analysis applied to this problem yields

$$St = \varphi Pr^{a} Np^{b} Re^{c} \qquad (7)$$

where the properties for evaluating the numbers are that of the fluid. The correlation by these relations and its subsequent comparison with pool boiling heat transfer coefficients show the desirability of wick boiling at low Q and Re_{ℓ}.

Experiments have been performed to show the temperature distribution, the boiling heat transfer coefficients, and the vapor temperature and pressure drop. The experimental set up consisted of a $\frac{3}{4}$ in. O.D. stainless steel pipe 24 to 36 in. long, with its interior wall covered by a wick material (Fig. 1). The wick used was a passivated, 100-mesh stainless steel screen with porosity varying from 0.65 to 0.94. Heat was added at the evaporator section by resistance heating. The entire pipe was insulated using polyurethane foam. The condenser section was kept in an ice bath. Temperatures were measured using a 24-point Daystrom recorder during transient and steady state operation.

The experiment was started by placing the tube in a vertical position and boiling off water in order to evacuate the tube of air and non-condensible gases. When there remained enough water to saturate

the wick and a very small excess, the tube was sealed and placed in a horizontal position. The heat input (700 to 6000 BTU/hr-ft^2) was adjusted to a predetermined value and condenser temperature controlled. The temperature of the vapor inside the heat pipe was monitored using a thermocouple embedded in an axial wire.

The wick boiling heat transfer coefficient is $h = Q/A(T_v - T_w)$ and the liquid mass flow rate is $W = Q/A\epsilon\lambda$. Selecting 0.6 as the exponent of the Prandtl number, to correspond to liquid heating literature, the dimensionless correlation is obtained as

$$St = 0.00051\ Pr^{.6}\ Re^{-1.43}\ Np^{.2} \qquad (8)$$

The above correlation is compared to previous data and shown in Fig. 2. It is clear from Fig. 2 and published pool boiling data[2] that higher results are obtained at low heat fluxes and lower numbers at higher heat fluxes. This leads to the confirmation of the idea that wick boiling is preferable at low heat fluxes. Presence of the wick material decreases turbulence near the surface, increases the effective surface area, and provides active sites for bubble formation, so that higher film coefficients at low heat flux are obtained compared to pool boiling. The correlated equation (8) affords some insight into the behavior of the heat pipe under varying conditions. Although the results were obtained for water, experimentation is continuing to determine applicability and performance of other working fluids.

The temperature distribution along the heat pipe is shown in Fig 3. The distribution is for steady state operation and indicates high thermal conductivity in the axial direction. The temperature difference of the vapor and the pressure differentials

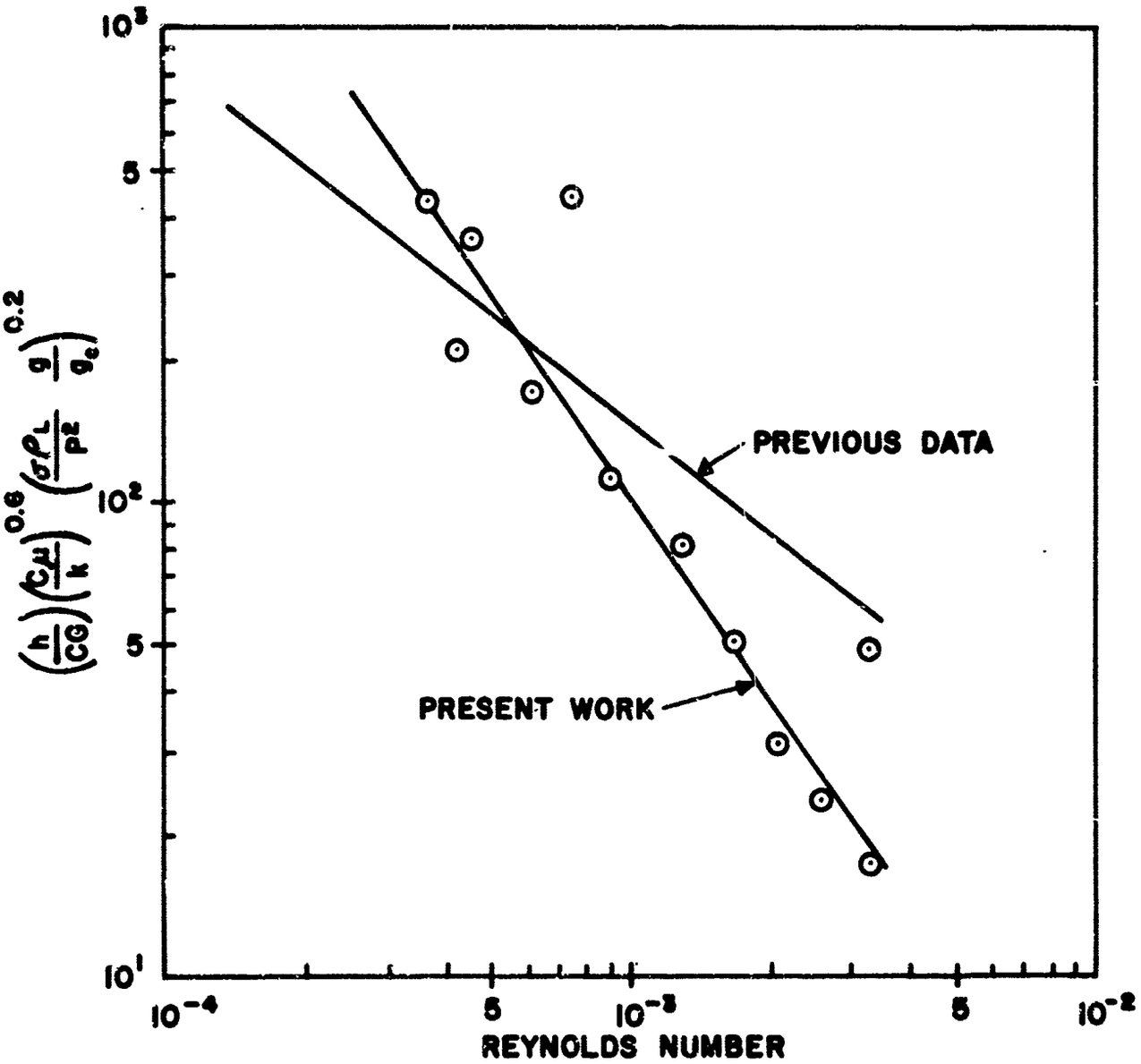

Fig. 2 WICK BOILING HEAT TRANSFER CORRELATION

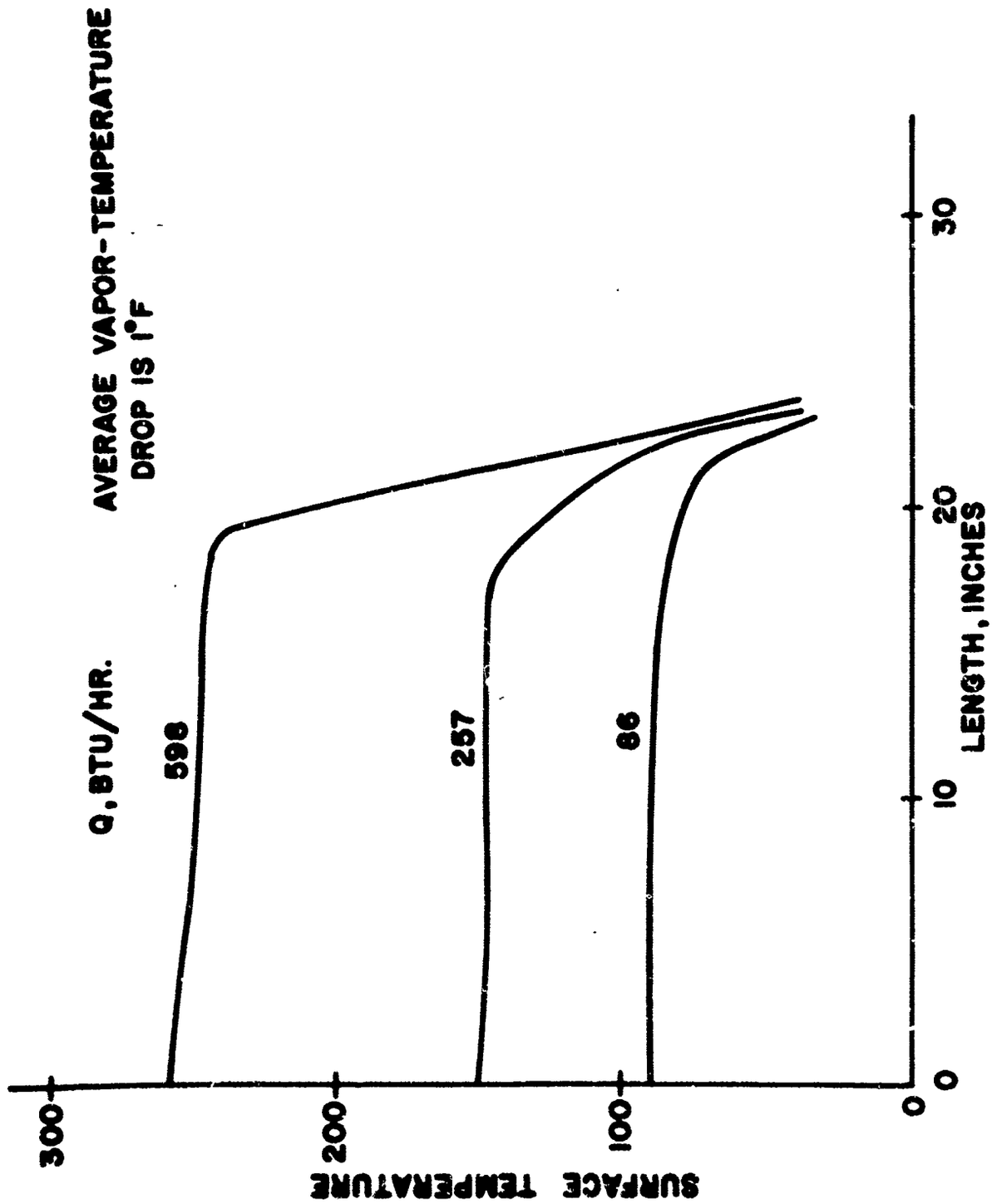

Fig. 3 TEMPERATURE DISTRIBUTION ON HEAT PIPE

were very small. Theoretical values obtained by using equations from Cotter have shown this. It is clear that very small temperature and pressure differences are required to supply the necessary driving forces for successful operation and that their magnitudes are not too important.

Fig. 4 is a plot of the heat pumped, in the axial direction off course, versus the difference between the average evaporator and condenser temperature. This again gives us some appreciation for the "effective thermal conductivity" that may be obtained.

Although the condenser section was held constant in our experiments, later experimentation included its variation. Thermal runaway occurred when large quantities of working fluid were trapped in the condenser thereby tending to deplete the supply in the evaporator section. The exact effect of other variables in thermal runaway has not yet been determined.

CONCLUSION

The foregoing results indicate the potential usefulness of the heat pipe. However, a method for actively controlling the device is needed. This work is being presently pursued at the Laboratory. It is clear that the type of wick and packing can be varied. The quantity of working fluid is not too important, provided that the entire wick is properly wetted with a very little excess. The choice of working fluid is dictated by temperature limitations and λ. For example, alcohol has low λ but also a lower freezing point than water.

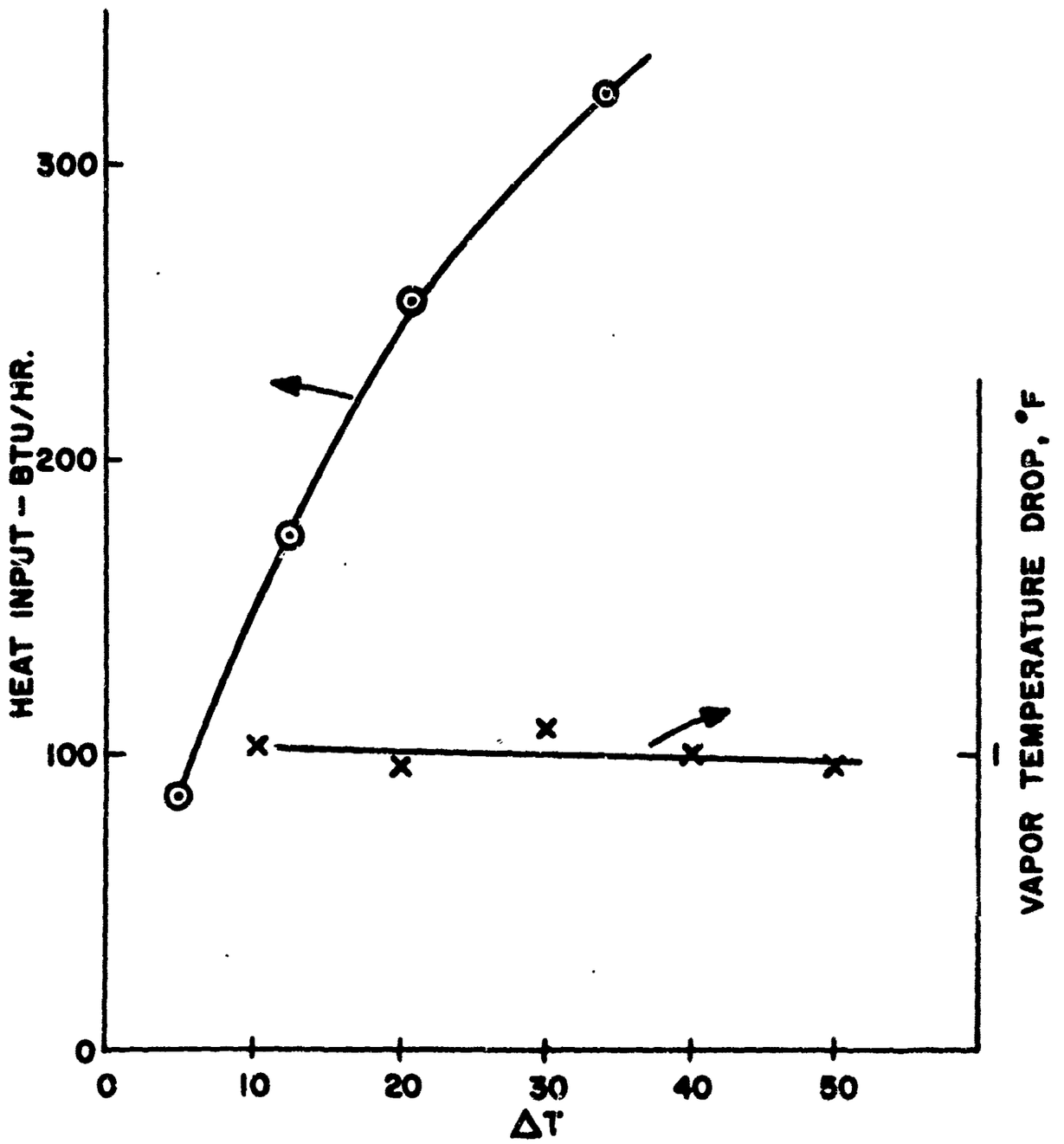

Fig. 4 HEAT FLOW VS. TEMPERATURE DIFFERENCE BETWEEN AVERAGE CONDENSER AND EVAPORATOR TEMPERATURE

The effect of other fluid properties may be deduced from equations (5), (6) and (8). There is no reason to believe that the pipe will operate at any preferred temperature although better efficiencies are obtained at low heat fluxes. The temperature and pressure differentials that act as driving forces are extremely small as compared to their absolute magnitudes of T and P.

REFERENCES

1. Grover, G.M., et al "Structures of Very High Thermal Conductance," Journal of Applied Physics, 35, 1900 (1964)

2. Allingham, W.D., and McEntire, J.A., "Determination of Boiling Coefficient for a Horizontal Tube in Water Saturated Wick Material," Transactions ASME, Series C, No. 1 (1961)

3. Cotter, T.P. "Theory of Heat Pipes," Los Alamos Scientific Laboratory, LA-3246-MS

4. Costello, C.P., and Redeker, E.R., "Boiling Heat Transfer and Maximum Heat Flux for a Surface with Coolant Supplied by Capillary Wicking," Chemical Engineering Progress Symposium Series, 59 (1963)

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SUBJECT: Heat Pipe Experiments II

REFERENCES: (1) Anand, D. K. and Dybbs, A. Z., "Heat Pipe Experiments," memo S1A-156-66 to R. E. Fischell

(2) Anand, D. K., "Heat Pipe Investigations," memo S1A-170-66 to W. H. Guier

Experiments at saturation temperatures of 106°F and 170°F have been previously reported.[1] The purpose of the present experiments is to operate the heat pipe at 68°F and observe the effect of various control techniques at this temperature. The working fluid is ethyl alcohol and the heat pipe is identical to that used in the previous experiments.

With appropriate selection of condenser parameters (area and surface temperature) the steady state heat pipe operation at 68°F is shown in Figure 1. The effect of varying heat input (from 30 to 60 watts) is shown in Figure 2. Note that the previous condenser parameters are maintained.

Figure 3 depicts the temperature distributions for varying condenser areas but a fixed condenser surface temperature. Under these conditions it is clear that the operating temperature becomes a direct function of the available condenser surface area for ordinary conduction.

The effect of introducing a control in vapor core is shown in Figure 4. The plot is for the valve in fully open position. It is clear that there is a pressure drop at the valve and this results in two heat pipe regimes operating at different temperatures. Note that the overall

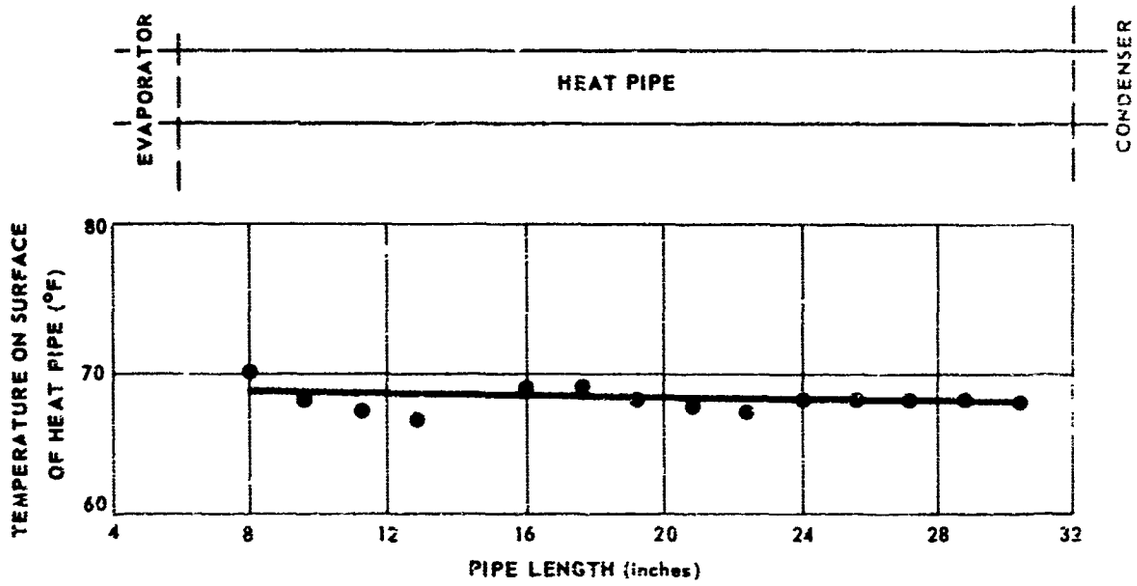

Fig. 1 HEAT PIPE OPERATING AT SATURATION TEMPERATURE OF 68° F (30 WATTS AND 12 INCH CONDENSER)

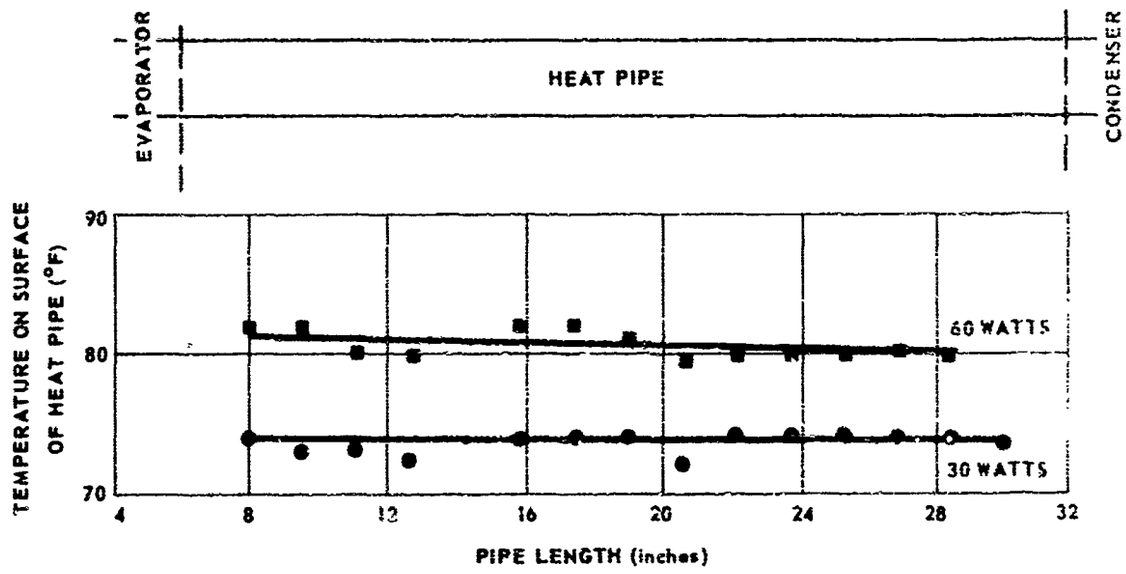

Fig. 2 EFFECT OF HEAT INPUT VARIATION ON HEAT PIPE WITH CONSTANT CONDENSER AREA (12 INCH) AND CONSTANT CONDENSER TEMPERATURE (54° F)

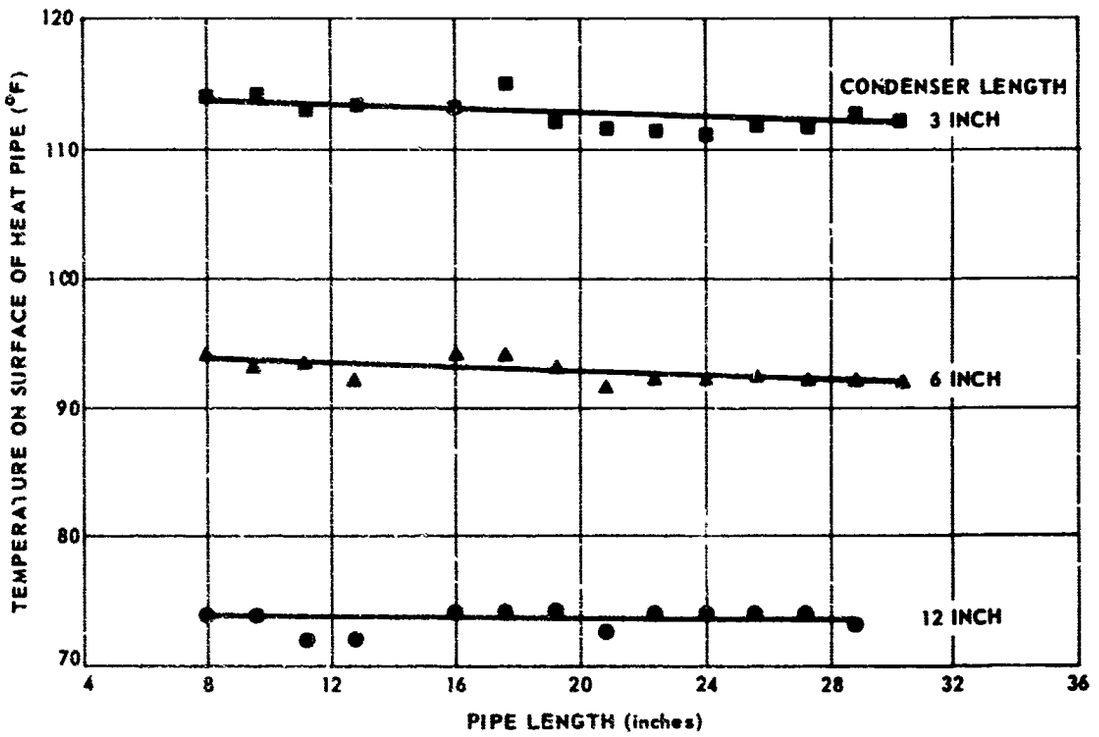

Fig. 3 VARIATION OF HEAT PIPE CONDENSER AREA WITH FIXED CONDENSER TEMPERATURE OF 54° F (30 WATTS IN ALL CASES)

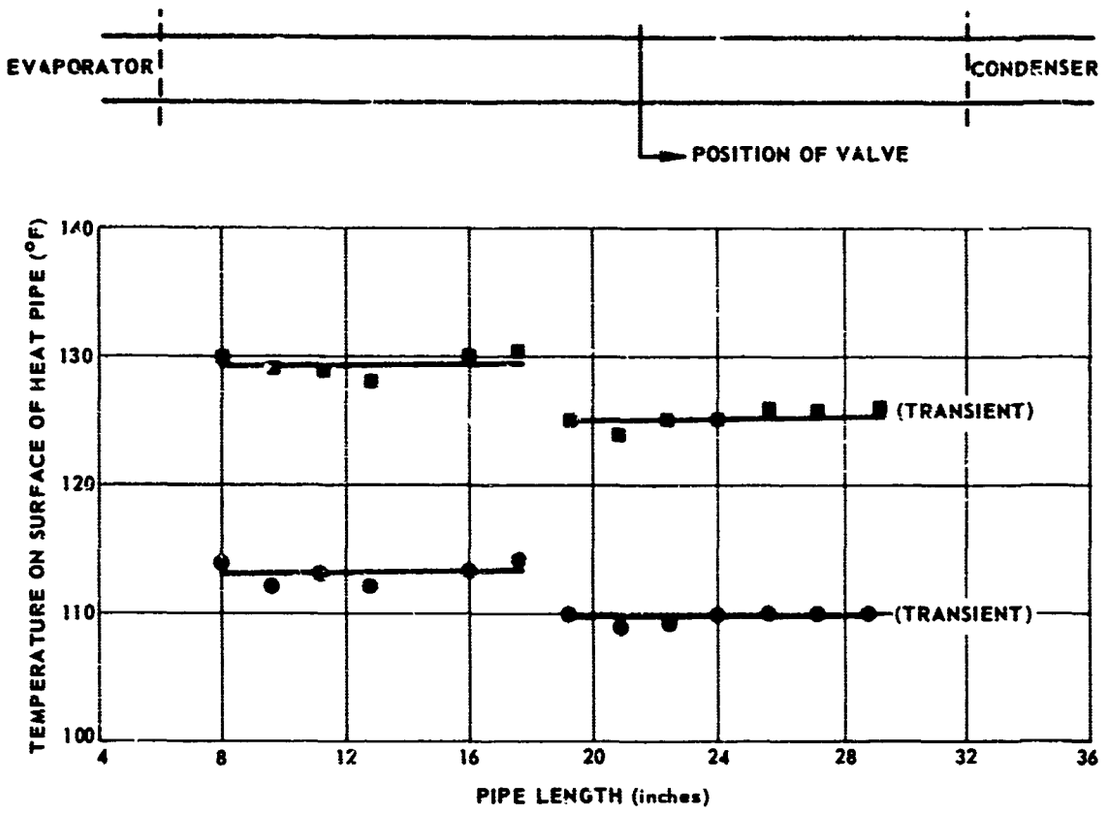

Fig. 4 HEAT PIPE OPERATION WITH VALVE OPEN SHOWING INCREASING TRANSIENT STAGES 60 MINUTES APART (30 WATTS AND 12 INCH CONDENSER LENGTH)

operation is at a temperature higher than that shown in Figure 1. This is so in spite of the fact that the condenser parameters are identical. The temperature distribution of Figure 4 is transient. The evaporator temperature could be decreased if the condenser surface temperature were decreased drastically.

From these and previous investigations[1,2], it is concluded that the control of condenser parameters is the most desirable control technique for satellite temperature control purposes.

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Subject: Heat Pipe Experiments III

References: (1) Anand, D. K. and A. Z. Dybbs, "Heat Pipe Experiments," memo S1A-156-66 to R. E. Fischell, dated 27 June 1966.

(2) Anand, D. K and A. Z. Dybbs, "Heat Pipe Experiment II," memo S1A-173-66 to R. E. Fischell, dated 29 July 1966.

(3) Private discussion with D. K. Anand, A. Z. Dybbs, R. E. Fischell, R. Harkness, and J. F. Smola

(4) Schlichting, H., Boundary Layer Theory, McGraw-Hill Book Co., First Edition, pp. 18-21, 26-27, 1955.

Introduction

Many different types of control devices and techniques have been tested in an effort to control the operation of the heat pipe.[1,2] Among these techniques was the interruption of vapor flow in the core of the heat pipe by use of mechanical valves and disks with various size holes. Recently a suggestion, concerning the valving technique, was made[3] that instead of cutting off large portions of the vapor core,[1] heat pipe control might be accomplished by only cutting off a small (20%) portion of this core. This proposal was made on the basis of experiments[2] performed with a valve whose actuating mechanism was a long rod connected to an inverted hemisphere as is shown in Figure 1. The purpose of the present experiments was thus to investigate the possibility of using small blockage of the vapor core of the heat pipe as a means of heat pipe control.

Experimental Apparatus

The heat pipe used in these experiments was of the same type as previously employed.[1,2] It consisted of a 42 in. long, 3/4 in. O.D. stainless steel tube with 6 layers of 1/100 in. thick stainless steel mesh. The evaporator section of the heat pipe was 6 in. long and a variable condenser length of 12 in. was used. Evaporation was achieved by resistance heating (30 watts) and condensation was accomplished by circulating cold water over the condenser length.

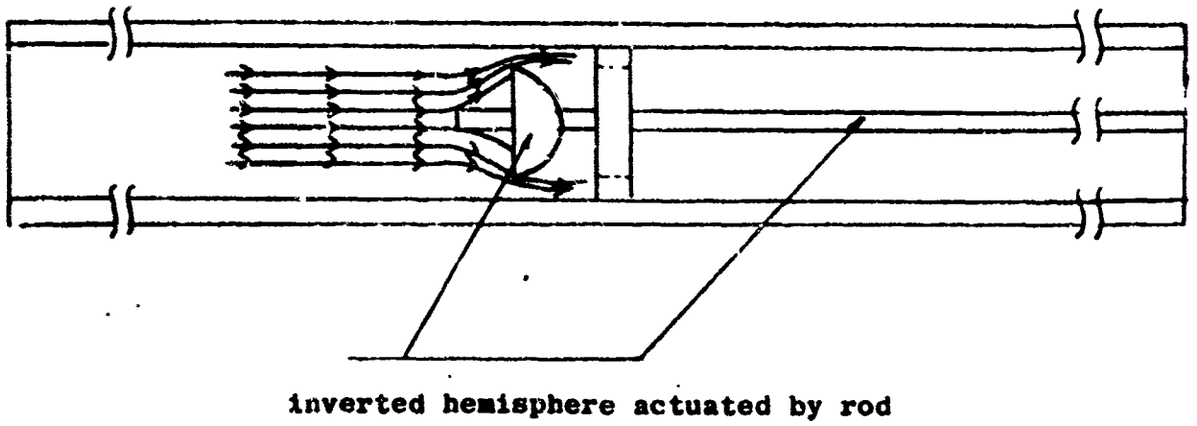

Figure 1. Valve Mechanism and Resulting Flow Pattern

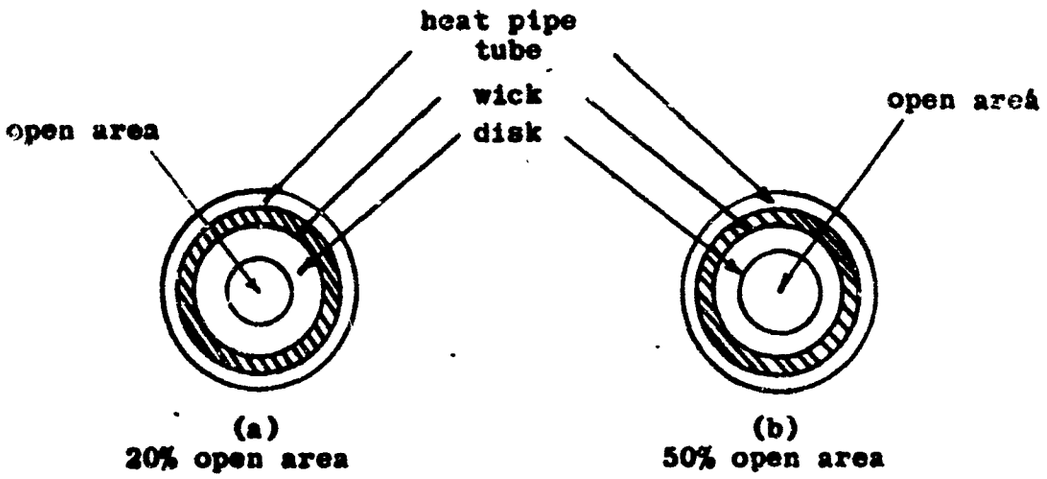

Figure 2. Cross-sectional View of Heat Pipe with Various Size Disks

Blockage of the vapor core of the heat pipe was achieved by employing various disks with symmetric circular holes. These disks were inserted at the entrance of the condenser section. The first disk (Figure 2a) blocked 80% of the available vapor core and the second (Figure 2b) about 50% of the core. The third case tested was with no blockage. In all cases the heat pipe working fluid was chemically pure ethyl alcohol and the condenser length was 12 in. or 6 in.

Discussion of Results

The results of vapor core blockage are shown in Figure 3. In this diagram the average temperature of the heat pipe regime--the length from the end of the evaporator to the entrance of the condenser--is plotted versus the percentage open area of the vapor core. The plots seem to indicate that at least for open areas greater than about 20%, there is an insignificant effect on the operation of the heat pipe. This result was anticipated by the author.[3]

It should be noted that the points at 20% open area vapor core were interpolated from past heat pipe experiments.[1] Although these experiments were operated at higher saturation temperatures, the previous results indicated that there was no perceptible difference in the operation[4] between the 20% open vapor core and the 100% open area. Thus, the present interpolation seems justifiable.

The difference in the results presented here and those of Ref. 2 for the valve can probably best be explained as follows: In the case of the valve the flow is in the direction of the inverted hemisphere (see Figure 1). This would cause a pressure drop similar to that observed for any type blunt body.[4] Since the heat pipe operates on or near the saturation pressure temperature curve the pressure drop would produce a corresponding temperature drop and hence the possibility of two heat pipe regimes as was obtained with the valve.

In conclusion one can say: That two heat pipe regimes are caused by the mechanisms necessary to operate a valve and not the valve itself; that the valve opening must be quite small before any type of control can be accomplished; and finally that if a mechanism for controlling a valve is introduced in the vapor cone, extreme care must be exercised to control pressure drops.

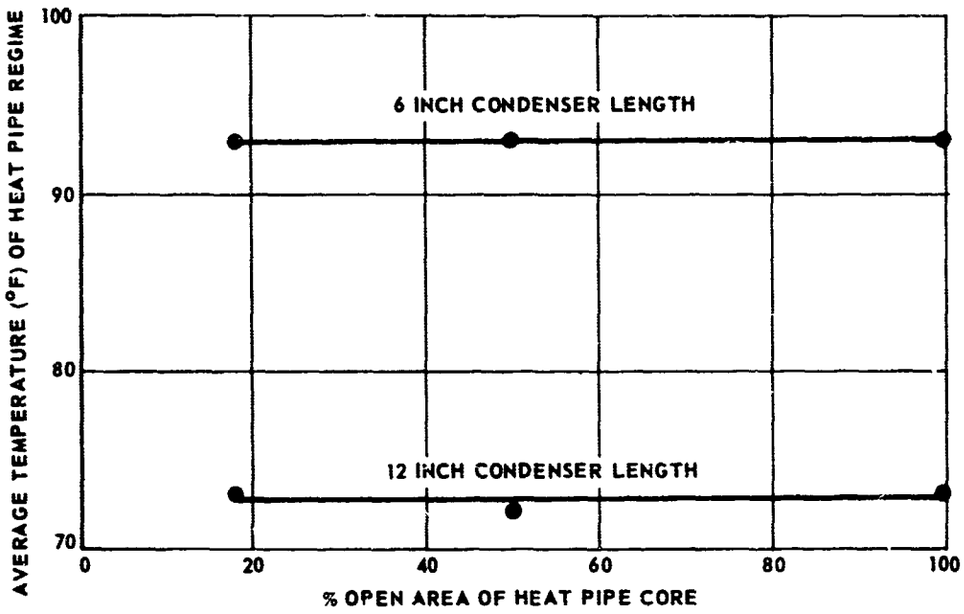

Fig. 3 EFFECT OF BLOCKAGE OF HEAT PIPE CORE ON HEAT PIPE OPERATION (ALL HEAT PIPES WERE OPERATED WITH 30 WATTS INPUT AND 54° F CONDENSER SURFACE TEMPERATURE)

SUBJECT: Heat Pipe Investigations

Experimental data[1,3,4] on sodium, ethyl alcohol, and distilled water heat pipes have been previously reported. Theoretical investigations have mostly dealt with either pressure gradients or wick boiling correlations. Cotter[2] has discussed multifluid and optimal heat pipes. The purpose of this paper is to show theoretical surface temperatures and performance as a function of the heat pipe parameters. The comparison between theoretical and experimental temperatures is very good. The operation of the heat pipe for satellite applications is seen to be constrained chiefly by condenser parameters.

Optimum Conditions:

The total heat transport in a heat pipe may increase as long as the capillary force can sustain the necessary circulation of the liquid in the wick. Furthermore the pressure differences at the liquid - vapor interface must be maintained in order to maintain a continual change of phase. For a capillary structure of pore radius r_c having a liquid with contact angle θ, the radius of the local meniscus is $r_c/\cos\theta$ and the local pressure difference must be

$$p_v(z) - p_\ell(z) \leq 2\,\gamma\cos\theta/r_c \qquad (1)$$

where the equality would yield maximum heat transport. For the case of uniform heat addition and removal, low radial Reynold number N_R, the viscous effects

predominate and the vapor pressure drop becomes that of Hagen Poiseuille flow through a cylindrical pipe. Substituting the pressure drops[4] Eq. (1) becomes

$$\frac{4\mu_v(\ell_e + L)\,Q}{\pi\rho_v\, r_v^4\,\lambda} + \frac{b\mu_\ell(\ell_e + L)\,Q}{2\pi(r_w^2 - r_v^2)\,\rho_\ell\, e\, r_c^2\,\lambda} - \frac{2\gamma\cos\theta}{r_c} = 0 \qquad (2)$$

for the requirement of maximum heat transport. If turbulence does occur, then vapor pressure must be determined from the Blasius equation:

$$\Delta p_v = -\,0.0655\,\mu_v^{\frac{1}{4}}\left(\frac{Q}{2\lambda\,\pi r_v}\right)^{7/4} z \qquad (3)$$

The onset of turbulence occurs when the Reynolds number based on axial flow (N_z) exceeds 1,000. In the problem encountered here $N_z < 1{,}000$ and the assumption of Hagen Poiseuille flow becomes quite adequate provided $N_R \ll 1$. Therefore, considering constant properties, the optimum value of r_c from Eq. (2) becomes

$$r_c = \frac{b\mu_\ell(\ell_e + L)\,Q}{2\pi(r_w^2 - r_v^2)\,\rho_\ell\, e\,\gamma\,\lambda\cos\theta} \qquad (4)$$

This substituted into Eq. (2) yields

$$\frac{c_1\, c_2}{r_v^4(r_w^2 - r_v^2)\,\gamma\cos\theta}\,Q^2 - \gamma\cos\theta = 0 \qquad (5)$$

where $c_1 = 4\mu_v(\ell_e + L)/\pi\,\rho_v\,\lambda$, $c_2 = b\mu_\ell(\ell_e + L)/2\pi\,\rho_\ell\, e\,\lambda$

Appendix E

It is clear that maximizing Q requires an optimum value for $r_v^4(r_w^2 - r_v^2)$. Since r_w is fixed, for any given case, then $\frac{\partial}{\partial r_v}(r_v^4 r_w^2 - r_v^6) = 0$ yields the constraint that $r_v^2/r_w^2 = \frac{2}{3}$. Substituting this and the expression for optimum r_c, the maximum heat transport becomes

$$Q = \frac{\pi r_w^3 \lambda \gamma \cos\theta}{z(L + \ell_e)} \left[\frac{2}{3} \frac{\rho_v \rho_\ell}{\mu_v \mu_\ell}\right]^{\frac{1}{2}} \tag{6}$$

If the operation of the heat pipe occurs at high N_R, the vapor pressure drop has been shown to be[7]

$$\frac{dp_v}{dz} = - \frac{s\dot{m}_v}{\rho_v r_v^4} \frac{d\dot{m}_v}{dz}$$

or

$$\Delta p_v = - \frac{(1 - \frac{4}{\pi^2}) Q^2}{8\rho_v r_v^4 \lambda^2} \tag{7}$$

where s = 1 for evaporation and $\frac{4}{\pi^2}$ for condensation.

Since Δp_v is not a function of the capillary pore size it plays no role in the determination of optimum r_c. Also since it is a function of r_v^4 (as Δp_v for Hagen Poiseville flow) the value of r_v^2/r_w^2 is still 2/3. Therefore, substituting Eqs. (4,7), $\frac{r_v^2}{r_w^2} = 2/3$, and Δp_ℓ from reference 4 into Eq. (1), the optimum Q becomes

$$Q = \frac{4\pi}{3} r_w^2 \lambda \left[\frac{\rho_\ell e \gamma^2 \cos^2\theta \rho_v}{b\mu_\ell(L + \ell_e)(\pi^2 - 4)}\right]^{\frac{1}{3}} \tag{8}$$

This then represents the maximum heat transport achievable for any specific case. It is, of course, a requirement that appropriate condenser areas be available for the maximum axial transport to leave by ordinary conduction.

For steady state conditions, the heat transfer through the condenser section must be

$$Q = 2\pi L k_e \Delta T / \ln \frac{r_o}{r_i} = c_3 L \tag{9}$$

under the assumption of uniform surface temperature. The constant c_3 is defined by the above equation and k_e is the effective thermal conductivity

Substituting for L in Eq. (6)

$$Q^2 + c_3 \ell_e Q - \frac{c_3}{3} r_w^3 \lambda \gamma \cos\theta [\frac{2}{3} \frac{\rho_v \rho_\ell}{\mu_v \mu_\ell} \frac{e}{b}]^{\frac{1}{2}} = 0$$

or

$$Q \text{ optimum} = [\tfrac{1}{3} c_3 r_w^3 \lambda \gamma \cos\theta (\frac{2}{3} \frac{\rho_v \rho_\ell}{\mu_v \mu_\ell} \frac{e}{b})^{\frac{1}{2}} - \tfrac{1}{4} c_3^2 \ell_e^2]^{\frac{1}{2}} - \frac{c_3 \ell_e}{2} \tag{10}$$

If $N_R \gg 1$, Eqs. (8) and (9) would yield optimum conditions. Depending upon the specific problem, constraints could be imposed on the heat flow or physical parameters, but often not on both.

The experimental temperature distribution reported[3] was for a heat pipe with the following parameters:

$(r_w - r_v) = 0.063$ inches

$\epsilon = 0.68$

$r_v/r_w = 0.815$

$r_c = 3.25 \times 10^{-3}$ inches

$Q = 30$ watts

$b = 15$

Fluid = Ethyl Alcohol

The optimum heat transport that could be achieved is 725 watts with a condenser area of 5 sq. ft., 0.1 effective conductivity, 60° contact angle, 1.5592×10^{-3} pound force per foot surface tension, and a temperature drop of 20° F It is clear that limiting conditions of the heat pipe, in satellite applications, may be primarily due to condenser parameters.

Surface Temperature

An energy balance on a differential element of the heat pipe wall and saturated wick under steady state conditions yields

$$- k_e A_c \frac{\partial t}{\partial z} = - k_e A_c \frac{\partial^2 t}{\partial z} dx + p_o h_o(t - t_c) dx$$

$$+ p_i h_i(t - t_i) dx - k_e A_c \frac{\partial t}{\partial z} \qquad (11)$$

or

$$k_e A_c \frac{\partial^2 t}{\partial z^2} - (p_o h_c + p_i h_i) t + (p_o h_o t_c + p_i h_i t_i) = 0$$

The temperature at $x = 0$ is t_v and at $x = L$ it is t_L. These temperatures refer to evaporator and condenser temperatures and are maintained as constant.

Substituting the boundary conditions and noting that t_i is the vapor temperature which is constant (0.75° F drop in a sodium pipe)[1] in the heat pipe regime, the temperature distribution becomes:

$$t = \left(t_v - \frac{E}{m^2}\right) \frac{1}{c} \left[e^{m(L-z)} - e^{-m(L-z)}\right] + \left(t_L - \frac{E}{m^2}\right) \frac{1}{c} \left[e^{mz} - e^{-mz}\right] + \frac{E}{m^2} \tag{12}$$

where:

$$E = \frac{p_o h_o t_o + p_i h_i t_i}{k_e A_c}$$

$$m^2 = \frac{p_o h_o + p_i h_i}{k_e A_c}$$

and k_e is the effective thermal conductivity between surface and vapor-liquid interface, h_o and h_i are outside and inside convective heat transfer coefficients, p_o and p_i are outside and inside surface perimeters, and A_c is the cross sectional area through which conductive heat transfer occurs. For insulated surfaces h_o is very small and the parameter of interest becomes $p_i h_i / k_e A_c$. This is identical to that used for studying performances of extended surfaces.[8] For a 3/4" stainless steel pipe the surface temperature is shown as a function of length with h_i as parameter in Figure 1. It is clear that higher values of h_i tends to decrease the effective condenser area. A higher value of h_i usually implies higher heat inputs which means that as Q is varied appropriate variations in the effective condenser area are desirable.

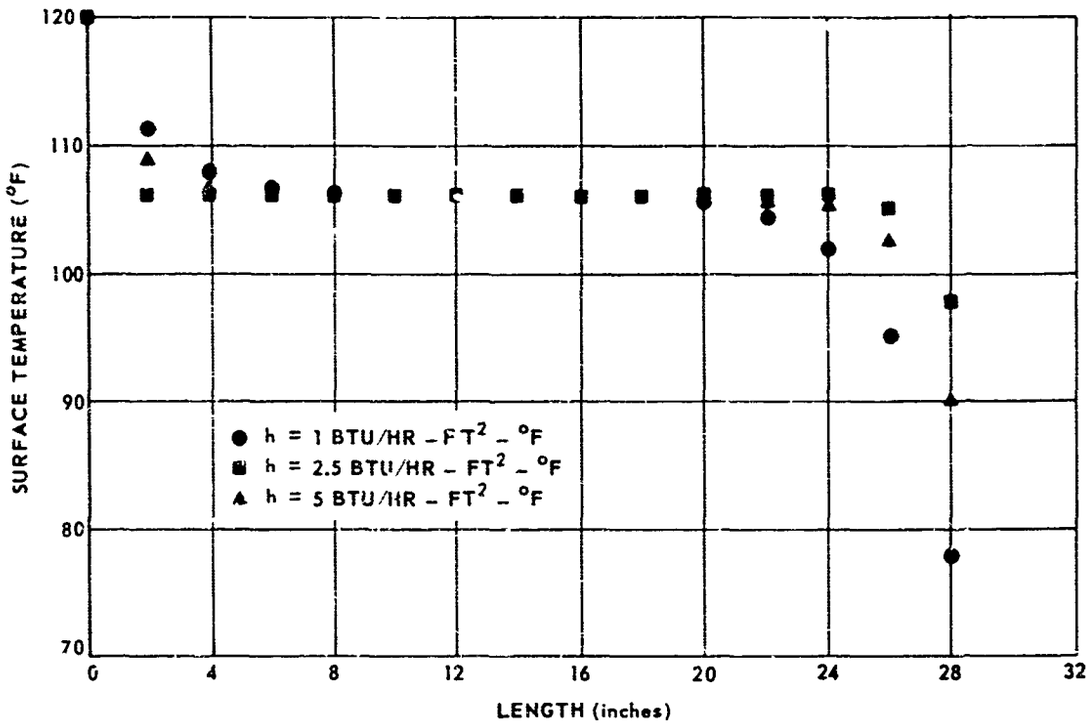

Fig. 1 COMPARISON OF THEORETICAL HEAT PIPE SURFACE TEMPERATURES WITH VARIATION IN HEAT TRANSFER COEFFICIENT h

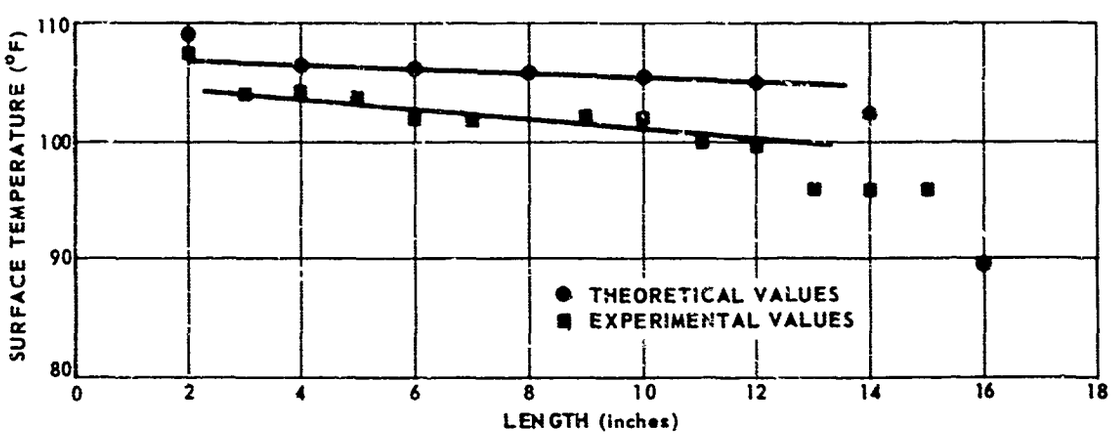

Fig. 2 COMPARISON OF THEORETICAL AND EXPERIMENTAL HEAT PIPE SURFACE TEMPERATURES (106° F SATURATION TEMPERATURE) h_i = 5 BTU/HR – FT² – °F)

Figure 2 shows a plot of the theoretical and experimental temperature[3] profiles in the heat regime for a saturation temperature of 106° F.

Both the optimum performance and the temperature distribution confirm the opinion that the control of condenser parameters is apparently the key to controlling the overall performance.

Nomenclature

A	area square feet
b	constant
e	porosity of wick
h	heat transfer coefficient of convection, BTU/hr sq. ft.
ℓ	length, ft.
L	length, ft.
N	Reynold number = $2\rho rv/\mu$
p	perimeter, ft.
p	pressure, lb_f/sq. ft.
Q	heat flow, BTU/hr.
r	radius, ft.
t	temperature °F
v	velocity, ft/sec
z	axial heat pipe coordinate
γ	surface tension, lb_f/ft
θ	contact angle of fluid, radians
λ	latent heat of vaporization, BTU/lb_m
μ	coefficient of viscosity, lb_f sec/ft^2
ρ	density, lb_m/ft^3.

Subscripts

c	wick pore
c	cross-sectional area
i	inside
ℓ	liquid
o	outside
R	radial coordinate
v	vapor
w	wick
z	axial coordinate.

References

1. Grover, G. M., T. P. Cotter, and G. F. Erickson, "Structure of very High Thermal Conductance," J. Appl. Physics, 35, 1990 (1964).

2. Cotter, T. P., "Theory of Heat Pipes," March 26, 1965, Los Alamos Laboratory, New Mexico.

3. Anand, D. K., A. Dybbs, "Heat Pipe Experiments," S1A-156-66.

4. Anand, D. K., "On Heat Pipe Performance," ARS, May 1966.

5. Allingham, W. D., and J. A. McEntire, "Determination of Boiling Coefficient for a Horizontal Tube in Water Saturated Wick Material," Trans. ASME, Ser. C, No. 1 (1961).

6. Grover, G. M., J. Bohdansky and C. A. Busse, "The use of a New Heat Removal System in Space Thermionic Power Supplies," Euratom.

7. Knight, B. W., and B. B. McInteer, "Laminar Incompressible Flow in Channels with Porous Walls," LADC 5308.

8. Jakob, M., "Heat Transfer," Vol. I, John Wiley and Sons.

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SUBJECT: Two Fluid Control

REFERENCES: 1. Anand, D. K., and Dybbs, A., S1A-156-66, dated 27 June 1966

2. "Mechanical Engineering," ASME, March 1966, page 71

Introduction

Three of the previous techniques reported,[1] for controlling the heat pipe, involved a fixed amount of working fluid in the main heat transfer loop. One technique required condenser flooding, but the fluid always stayed within the main heat pipe. The technique to be discussed here involves a small reservoir and a secondary fluid for control purposes.

The System

The system containing the heat pipe, isothermal enclosure, control fluid and reservoir is shown in Figure 1. It is a requirement that the P-T curve of the working fluid be steeper than that of the control fluid. The point of intersection, the cross over temperature T_0, represents the temperature desired of the isothermal enclosure.[2] This is shown in Figure 2.

Theory of Control

Assume that the temperature of the enclosure is T_2. Then the pressure of the fluid C will be P_{2C} and this will be equal to the pressure P_0 of the working fluid. The temperature of the working vapor is T_2 (ideally) and its pressure is P_{2W}, assuming there is no super heat. The pressure difference $(P_{2W} - P_{2C})$ represents the pressure necessary to drive the vapor to the

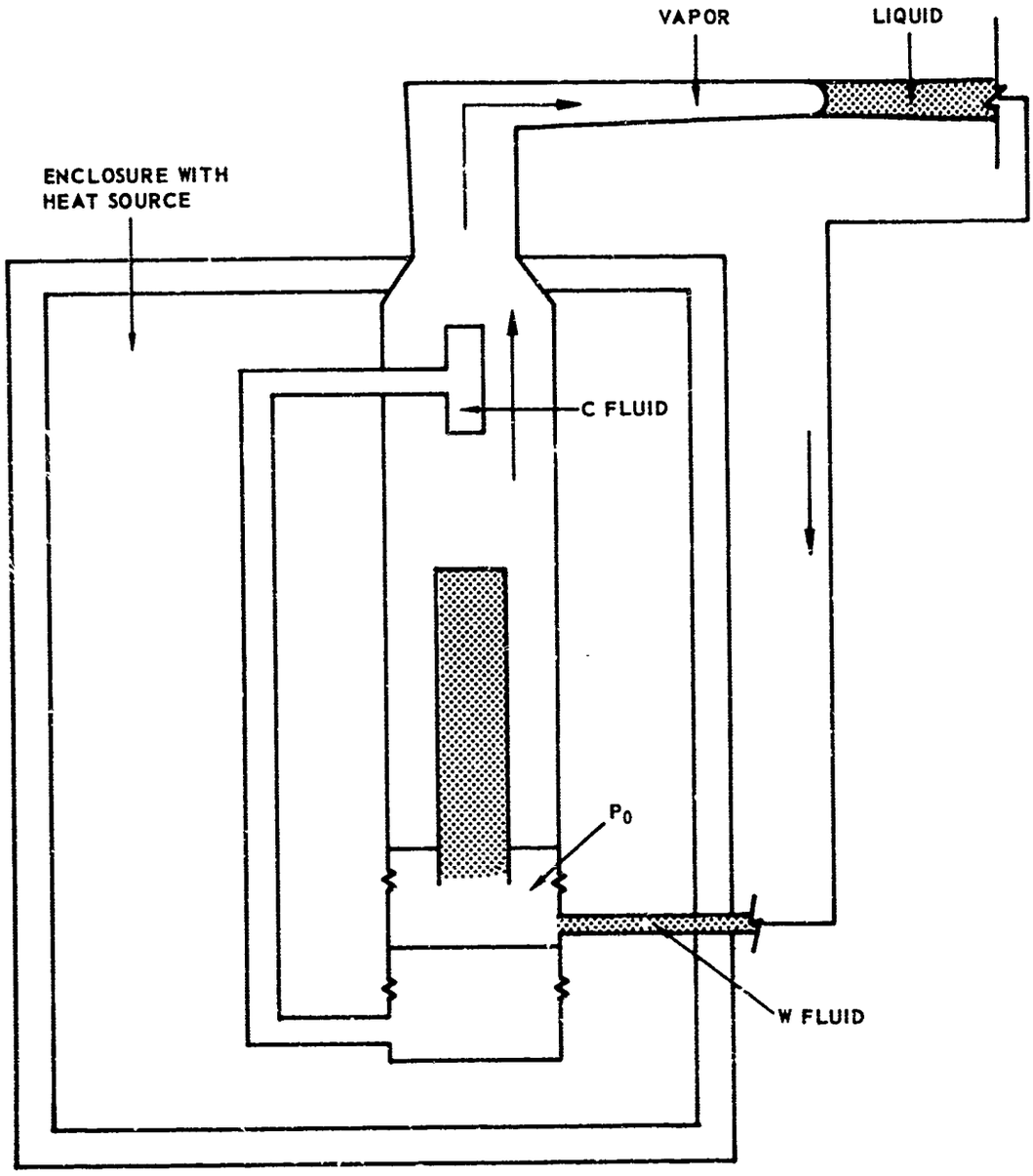

Fig. 1 SYSTEM SHOWING THE HEAT TRANSFER LOOP (no scale)

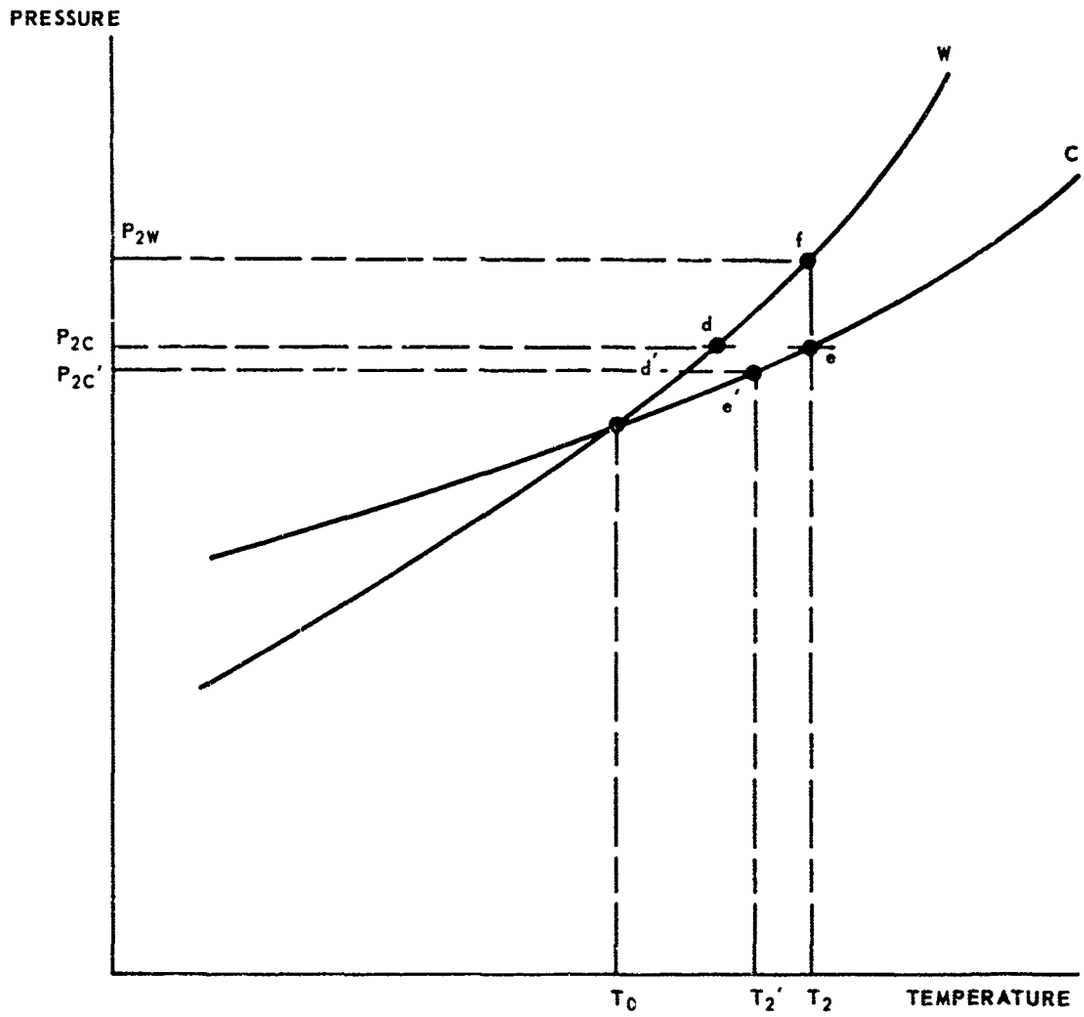

Fig. 2 P–T RELATION OF CONTROL AND WORKING FLUIDS

condenser and enter the reservoir in liquid phase to repeat the cycle. Now this cycle repeats until T_2 decreases. As T_2 decreases the triangle 'fed' also moves towards T_0 and the pressure difference (represented by e and f) decreases. Essentially when the enclosure reaches T_0, the triangle vanishes and there is no pressure drop to sustain the flow. When this happens, the fluid meniscus in the condenser will move until the condenser is completely full of the working fluid. The heat cycle is now stopped.

The above explanation assumes that the temperature of C and vapor is equal. In reality C will lag the vapor temperature so that when vapor is at T_2 the control fluid will be at $T_{2'}$. Since the pressure corresponding to $T_{2'}$ is $P_{2C'}$ the pressure drop becomes $(P_{2W} - P_{2C'})$ which is greater than before. Obviously this is desirable.

Below T_0, the control pressure is greater than vapor pressure and the condenser stays flooded.

Comments

In comparing this to reference (1) it is clear that the above technique is similar to condenser flooding.

The above control however is slightly different from that reported in reference (2).

This was discussed with the author of reference (2) and he agrees. It seems that the overall technique merits close attention. A number of test have been conducted by Lewis Research Center in order to substantiate

the above ideas. It appears that experimentation has confirmed theoretically suspected results.

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SUBJECT: Heat Pipe Experiments

Introduction

The operation of a heat pipe has been demonstrated before at elevated temperatures with sodium as the working fluid.[1] Later work reported, on the performance of a heat pipe, pertained to the use of distilled water as the working fluid.[2] However, for the purposes of using a heat pipe for satellites, a working fluid having lower freezing points is desirable. The present work deals with heat pipes using ethyl alcohol as the fluid. The major effort was directed at analyzing various methods to control the heat pipe.

Control Techniques

The various techniques may be broadly classified into the following categories:

1. Interruption of vapor-flow through the core
2. Interruption of fluid-flow through the annulus
3. Control of condenser parameters
   a. Variation of condenser area
   b. Condenser flooding
4. Introduction of non-condensable gases
5. Control of the pressure (thereby temperature of evaporation) of the working fluid.

The actual mechanism may involve the use of one or more of the above techniques.

Experiment

The heat pipes used for all the experiments consisted of 3/4" stainless steel tube with 6 layers of stainless steel wick, each 1/100" thick. Heat was supplied by resistance heating over 2" of the evaporator length. The sink consisted of an ice bath in which 4" of the pipe was immersed. In later experimentation a wick was used to cool the condensing section.

The working fluid was ethyl alcohol (C_2H_5OH) and enough was introduced to completely saturate the wick with a little excess. A heat pipe is shown in Figure 1.

Interruption of vapor-flow was achieved by using three plugs of varying openings. One plug had no opening, thereby simulating a perfectly closed valve.

Interruption of fluid flow is achieved by introducing a 2" discontinuity, in the wick, located about 4" from the condenser section. The effect of a wick discontinuity is negated by tilting the pipe by 3°. The tilt introduces a gravitational term that insures the return of the liquid to the source even in the absence of capillary pumping.

The third technique is self explanatory. The non-condensable gas method utilized air. The fifth technique will be described in detail in a separate memorandum.

Results

Figure 2 shows the steady state heat pipe regime with saturation temperature (ST) of 160°F. The surface (heat pipe) temperature drop is about 4°F.

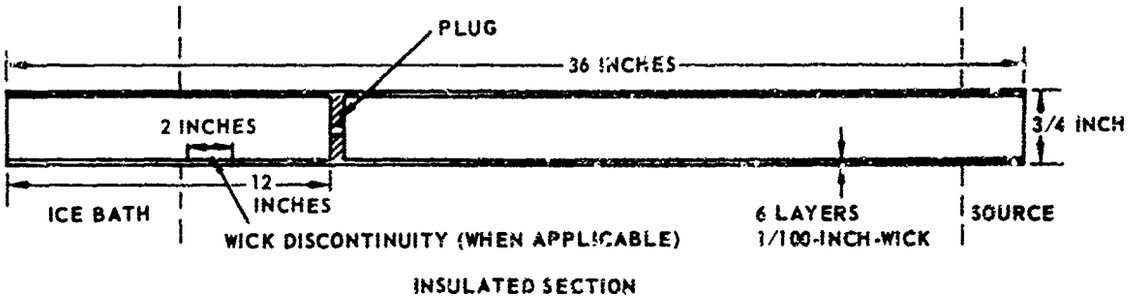

Fig. 1 HEAT PIPE WITH VARIOUS MODIFICATIONS WHEN APPLICABLE

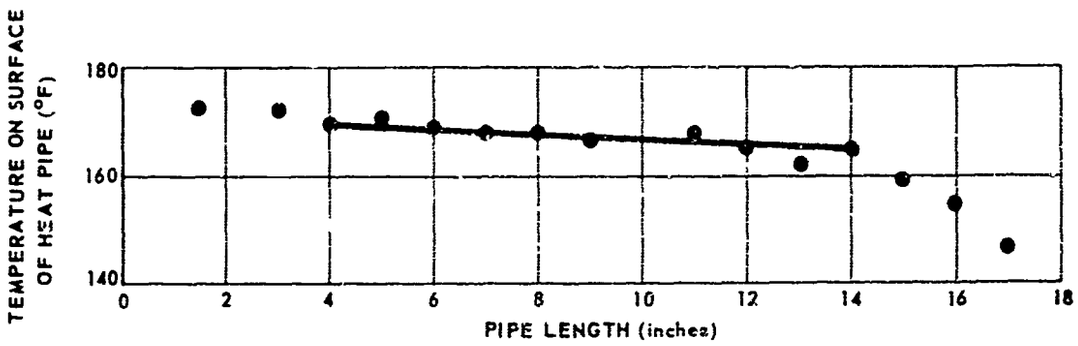

Fig. 2 NORMAL HEAT PIPE REGIME WITH NO CONTROL;
ETHYL ALCOHOL AS WORKING FLUID (SATURATION TEMPERATURE: 173° F)

Figures 3 and 4 show the effect of interrupting liquid flow to the evaporator. The dotted lines represent transient readings obtained ten minutes after eliminating the 3° tilt. Figure 4 was obtained at higher saturation temperatures than Figure 3. It is clear that fluid interruption does indeed choke the pipe.

The effect of vapor interruption is depicted in Figures 5,6, and 7. Heat pipe regime is obtained with a plug having 1/4" D hole. The choking effect begins to prevail when the opening is 1/10". The decrease in area represents a sufficient pressure drop to induce condensation during vapor passage. The pressure drop coupled with a lower plug temperature (due to radiation exchange) tends to cause a heat pipe regime between the evaporator and plug section. Figure 7 exhibits a heat pipe regime confined to the length between plug and source. This plug has no hole. Steady state is not achieved and the vapor tends to superheat. The liquid collects in the condensing section.

Figure 8 shows the heat pipe regime in steady state conditions. Excess fluid is introduced and this is driven towards and collects at the condenser section thereby decreasing the effective area. Since the condenser area is insufficient, all the heat cannot be transferred by ordinary conduction thereby decreasing the heat pipe effectiveness.

The non-condensable gas (air) has a similar effect as fluid flooding. This is shown in Figure 9.

The majority of the above experiments were conducted at or near 173°F ST and partial evacuation was achieved by boiling. Figure 10 shows that heat pipe regime obtained at lower ST where air was evacuated by a vacuum pump. Figure 11 simply illustrates "travel" up the p-T curve.

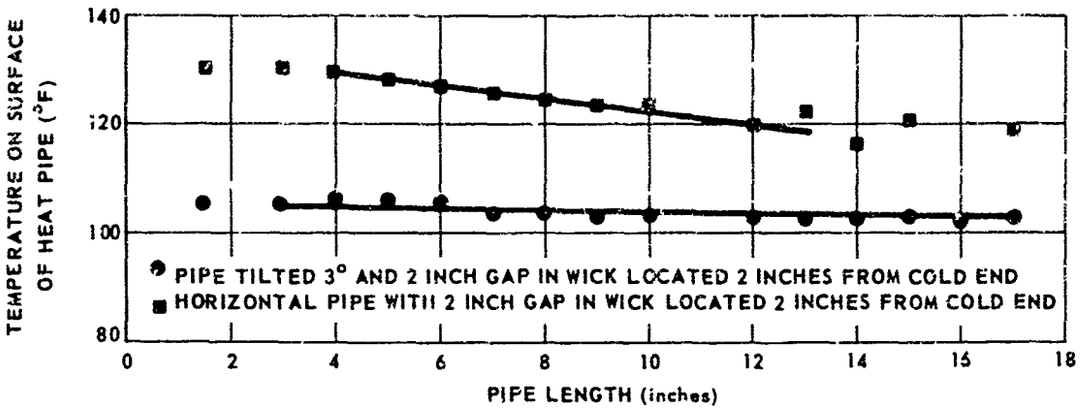

Fig. 3 COMPARISON OF TEMPERATURE DISTRIBUTION WITH EFFECT OF DISCONTINUOUS WICK, 100° – 140° F

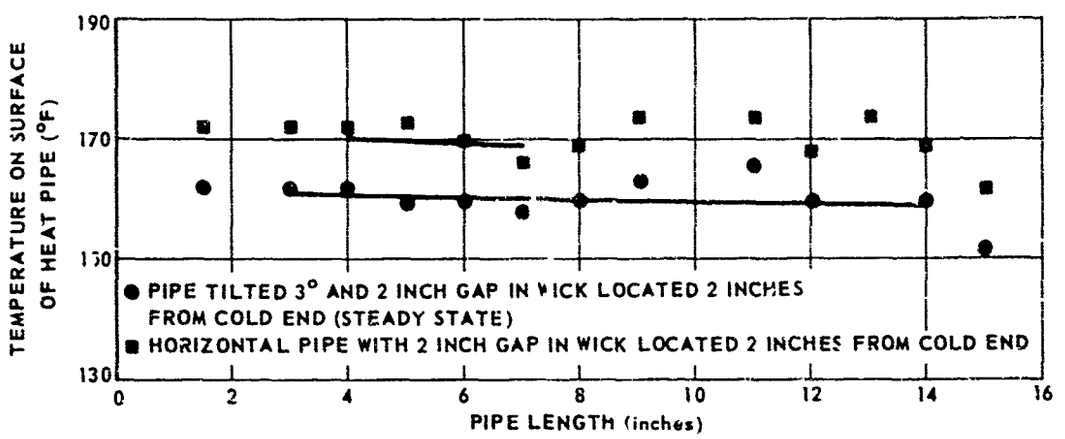

Fig. 4 COMPARISON OF TEMPERATURE DISTRIBUTION WITH EFFECT OF DISCONTINUOUS WICK, 150° – 190° F

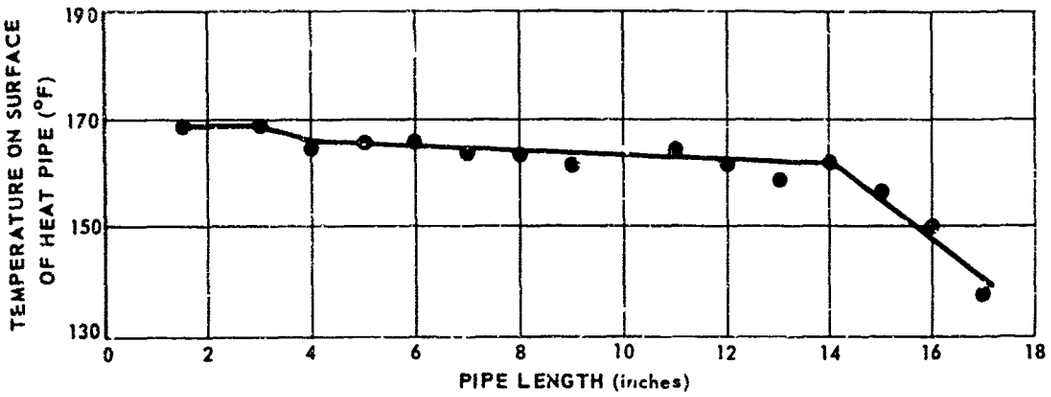

Fig. 5 INTERRUPTION OF VAPOR FLOW (PLUG WITH 1/4-INCH-DIAMETER HOLE)

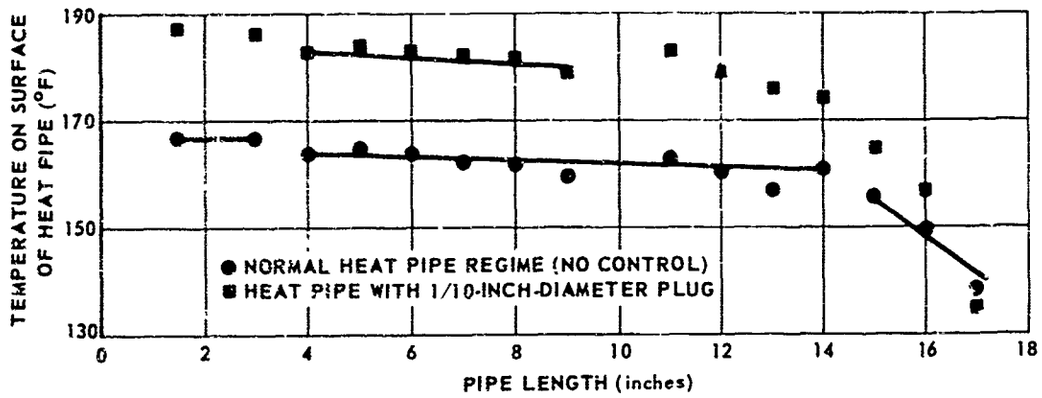

Fig. 6 INTERRUPTION OF VAPOR FLOW (PLUG WITH 1/10-INCH-DIAMETER HOLE)

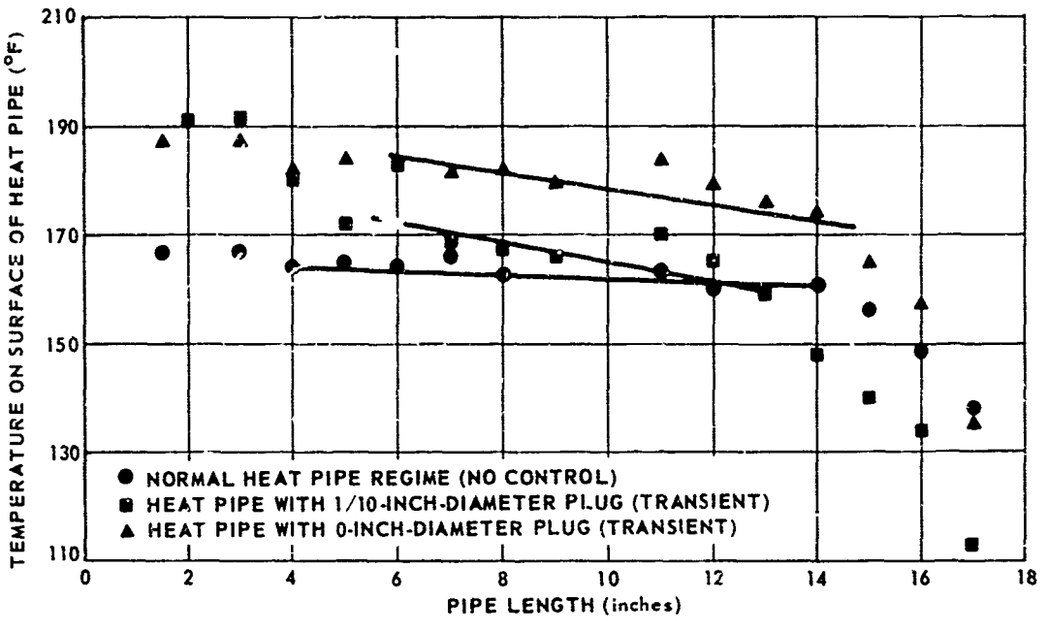

Fig. 7 INTERRUPTION OF VAPOR FLOW (PLUG WITH NO HOLE)

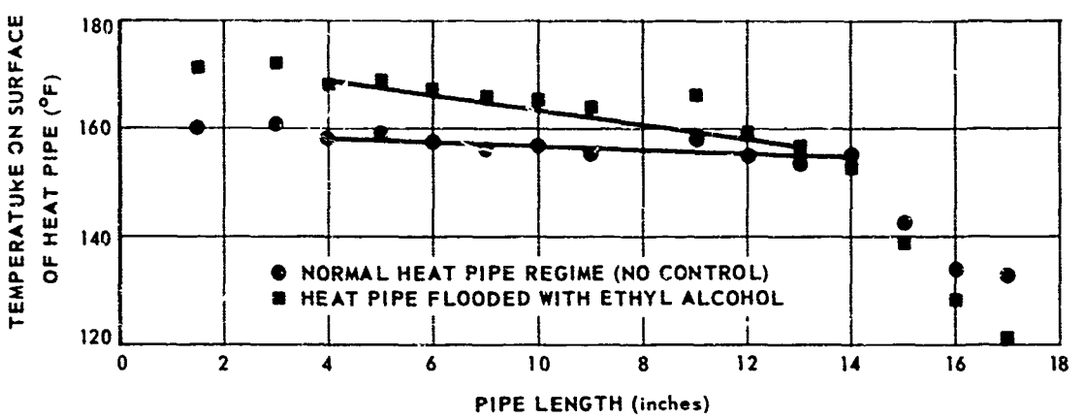

Fig. 8 HEAT PIPE WITH EFFECTS OF FLUID FLOODING

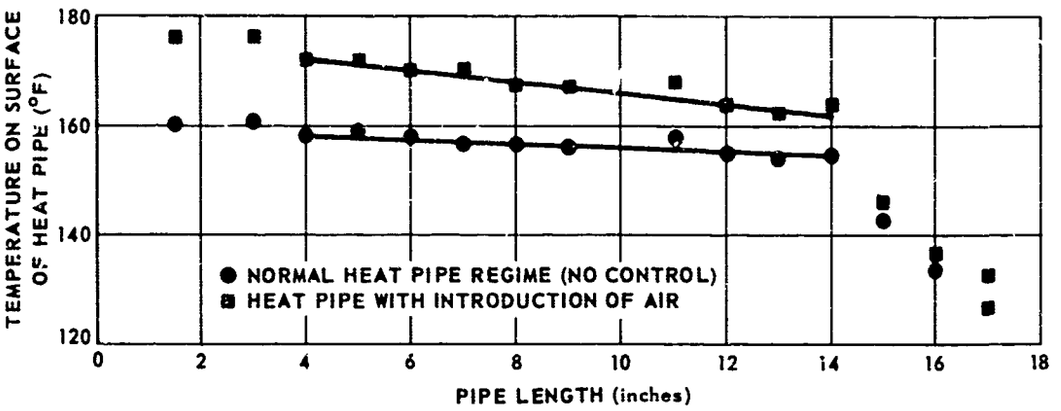

Fig. 9 HEAT PIPE WITH INTRODUCTION OF AIR

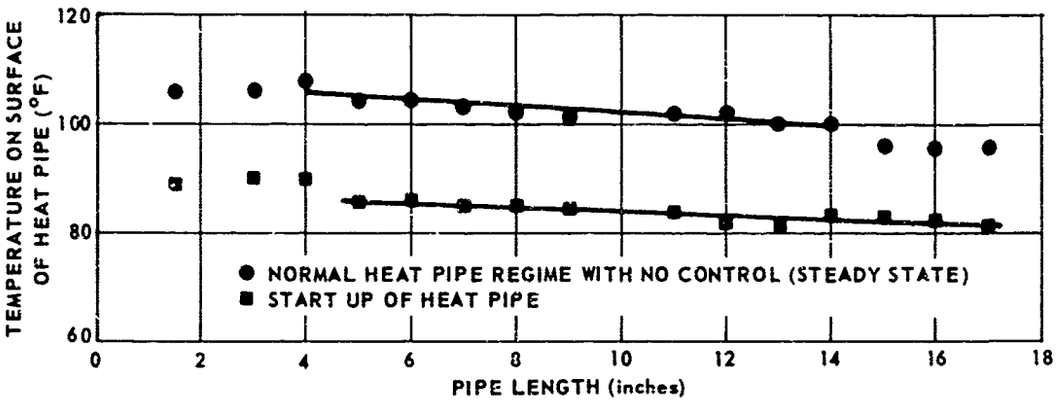

Fig. 10 HEAT PIPE OPERATION AT LOW SATURATION TEMPERATURE (106° F)

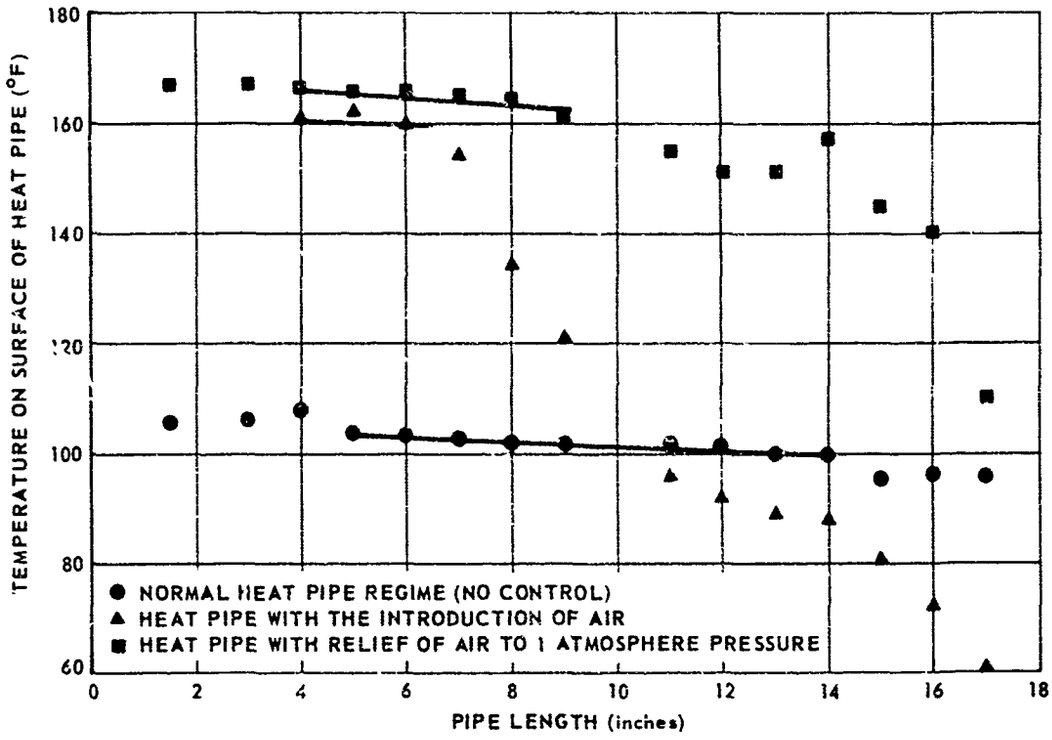

Fig. 11 HEAT PIPE WITH THE INTRODUCTION OF AIR AFTER BOILING AT LOW SATURATION TEMPERATURE (106° F)

Comments

Generally the control techniques were satisfactory. The choking of the pipe when the vapor regime was interrupted, with a plug having a 1/10" D hole, suggests that valves with small openings can cause pressure drops and consequently short circuiting. The effect of condenser parameter variation was most satisfactory. This technique comes closest to stopping the heat pipe. Introduction of air as a non-condensable is an inferior technique owing to the existence of water vapor which acts as an impurity if sufficient quantity is introduced.

Since the plugs were aluminum and the heat pipe was stainless steel, an electrolytic reaction would result with sustained heat pipe operation.

It is felt that the evaporator and condenser walls must be of high thermal conductivity whereas the middle section need not be. High thermal conductivity of the wick is also very desirable.

It was noticed that a loosely packed wick can cause alcohol vapor bubbles to be trapped between pipe wall and wicks. This causes a region of high temperature to travel up to the evaporator and collapse. There is however no particular significance to this phenomena.

References

1. Grover, G. M., T. P. Cotter, and G. F. Erickson, "Structures of very high thermal conductance," J. Appl. Phy. 35, 1900 (1964).

2. Anand, D. K., "On heat pipe performance," JSR, May 1966.

3. Wyatt, T., SDO-1389.

4. Swet, C. J., SDO-1167.

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Subject: Partial Analysis of the Heat Pipe

Reference: (a) APL/JHU Memo SDO-1154 dtd 29 March 1965

In order to assist in the work on heat pipes and to allay any lingering doubts regarding the certainty of operation of the heat pipe at a predictable level in the orbital condition a simplified analysis of one aspect is presented herein for consideration and comment.

It appears probable that one of the major uncertainties in the practical design of a heat pipe and also the principal source of doubt concerning the operation of the heat pipe in the so-called "zero-g" environment of orbital motion is the nature of the capillary flow in the device. Assume a capillary tube in static equilibrium as depicted in Figure 1. The pressures acting on the liquid in the tube are the capillary pressure, $\frac{4\tau \cos \theta}{D}$, the gas pressures P_1 and P_2, and the gravitational head, $h_s\, a\, \rho_\ell$, using the following symbols:

τ is the surface tension of the liquid

θ is the angle of contact at the edge of the meniscus between the liquid and the wall of the tube

D is the inside diameter of the tube

P_1 and P_2 are the gas or atmospheric pressures at the points noted

h_s is the liquid head under static equilibrium

a is the acceleration, gravitational or otherwise, against which the tube is working

ρ_ℓ is the density of the liquid

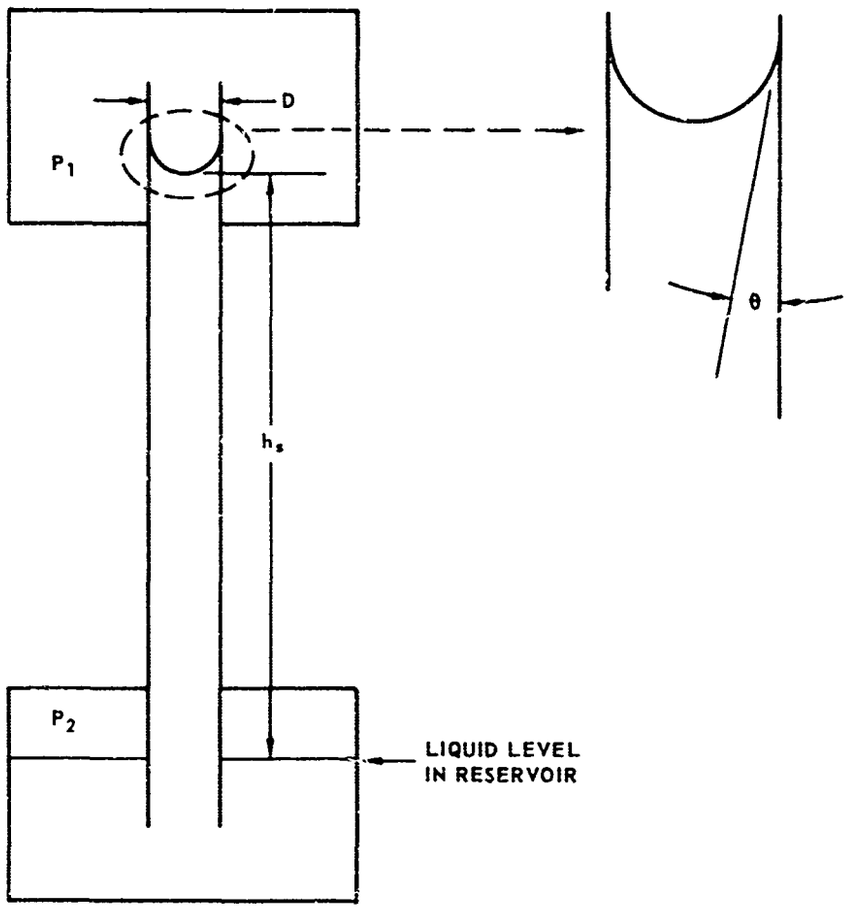

Fig. 1 CAPILLARY TUBE IN STATIC EQUILIBRIUM

With liquid-solid combinations wherein wetting occurs θ is less than 90°, as illustrated, and if the wetting is particularly good, as in the case of water and glass, $\theta \approx 0$. Thus the static pressure equilibrium of a selected case can be written

$$\frac{4\tau}{D} = h_s \, a \, \rho_\ell + (P_1 - P_2) \qquad (1)$$

The capillary tube is capable, under appropriate conditions, of delivering a sustained flow of liquid to some sort of a removal process, such as evaporation at the meniscus as shown in Figure 2. An "appropriate condition" in the context of the heat pipe is a value of h_f less than h_s. Assume that the sketch illustrates dynamic equilibrium, i.e., the flow rate and the evaporation rate are equal. Because of the low velocity of flow and the large length-to-diameter ratio of the tube the viscous flow relationship commonly attributed to Poiseuille is applicable

$$\partial\rho = \frac{128 \, \mu \, L \, \dot{V}}{\pi \, D^4 \, g} \qquad (2)$$

where

$\partial\rho$ is the pressure drop in the liquid due to fluid flow

μ is the absolute viscosity of the liquid

L is the length of the tube

$\dot{V}$ is the volumetric flow rate

D is the inside diameter of the capillary tube, or the "equivalent tube diameter" of a material which does not actually have discrete, measurable tubes. It is assumed that the actual or equivalent tube diameter is the same for both capillary and Poiseuille considerations.

g is the gravitational constant and is employed for consistency of units

and

h_f is the liquid head under a flow condition

The dynamic equilibrium can be written in terms of pressures as

$$\frac{4\tau}{D} = h_f \, \rho_\ell \, a + (P_1 - P_2) + \frac{128 \, \mu \, L \, \dot{V}}{\pi \, D^4 \, g} \qquad (3)$$

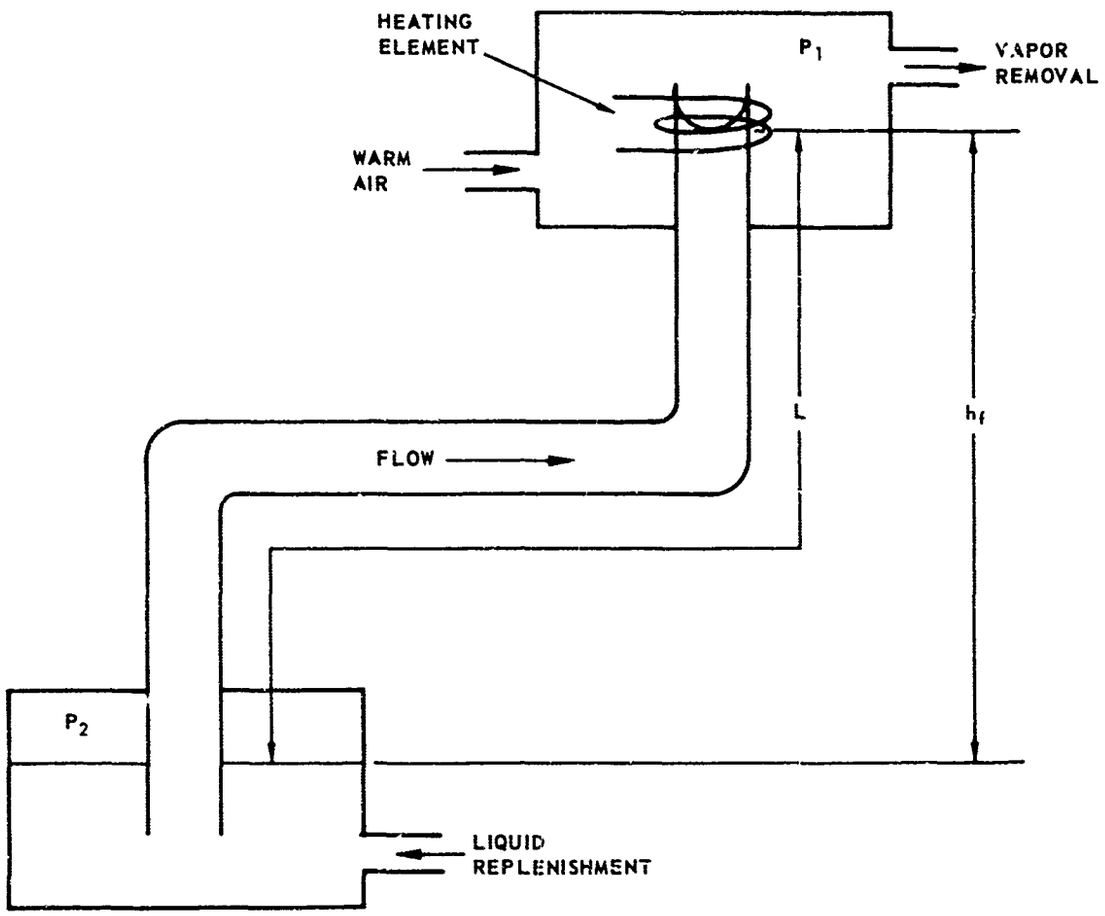

Fig. 2 EVAPORATION AT THE MENISCUS

It is apparent that the smaller the tube the greater is the capillary pressure tending to produce flow; on the other hand, the larger the tube the greater is the flow rate for a given available pressure.

In order to examine the counterplay of these contradictory trends we shall hold all of the variables in the design constant except tube diameter, D. Then we can substitute in Equation (3)

$$k = h_f \rho_\ell a + (P_1 - P_2)$$

and rearrange, getting

$$\dot{V} = \frac{\pi g D^4}{128 \mu L} \left[\frac{4\tau}{D} - k \right] \qquad (4)$$

Differentiating with respect to D and setting $\frac{d\dot{V}}{dD} = 0$ we find that the maximum value of $\dot{V}$ occurs when

$$D = \frac{3\tau}{k} \qquad (5)$$

Equation (5) is the basis for the designer to specify the equivalent tube diameter which the capillary material should have for his application and equation (1) provides means for him to determine experimentally (by observation of the static head) the equivalent tube diameter of various materials in order to select one meeting the requirement.

The other basic characteristic of the capillary material is the number of equivalent tubes in a unit cross section of the material. One way of making this determination might be to observe the volume of flow during an interval of time through a relatively short length (so that $\dot{V}$ would be large) with accelerated evaporation induced at the discharge end as shown in Figure 2 and then to calculate the number of multiples of $\dot{V}$ (from Equation 4) present. In the case of a very uniform material, such as porous ceramic or porous sintered metal (but not wire mesh or woven materials) wherein there are no voids or passages larger than the equivalent tube diameter a simpler procedure might be to apply a pressure differential across a sample and, using Equation 2, determine the number of multiples of $\dot{V}$.

Applying the foregoing reasoning to a heat tube operating at an input temperature of 20° C and using water as the working fluid we employ the following physical constants and assumptions:

τ = 72.75 dynes/cm

g = 981 cm/sec^2

a = g and 0

ρ_ℓ = 1.0 gm/cm^3

μ = 0.01002 poise (gm/sec cm)

Since the heat input end is at 20° C, P_1 = 17.535 mm Hg or 23,400 dynes/cm^2.

We shall assume a temperature drop across the heat tube of 0.2° C, which is probably excessively large, but show later that this assumption is not critical in the gravitational case.

Then

P_2 = 17.319 mm Hg or 23,100 dynes/cm^2

For the application suggested in Reference (a) the geometry is approximately as illustrated in Figure 3, which is a marked-up version of an earlier sketch employed in Reference (a). Although we are probably justified in designing for a desired flow rate to the midpoint of the battery tube we shall employ both that tube length and head and also the length and head to the top of the battery tube, the latter for conservatism. Then

Case I		and Case II	
h_f	= 15.9 cm	h_f	= 24.8 cm
L	= 17.8 cm	L	= 26.6 cm

Since we are interested in both the performance of the capillary under an environment of 1 g, e.g., in a laboratory thermal-vacuum test chamber, and also under an environment of zero-g in orbit we shall first solve equation (5) for each set of conditions, and then equation (1) and (4). The results are tabulated below.

	1 g environment*			0 g environment		
	D cm	$\dot{V}$ cc/sec	h_s cm	D cm	$\dot{V}$ cc/sec	h_s** cm
Case I	.0137	2.52×10^{-2}	21.6	.7275	3.78×10^{3}	.41
Case II	.0088	4.5×10^{-3}	33.7	.7275	2.53×10^{3}	.41

* i.e., working against 1 g

** not applicable in 0 g, but the value quoted is for the optimum value of D for 0 g as observed under 1 g.

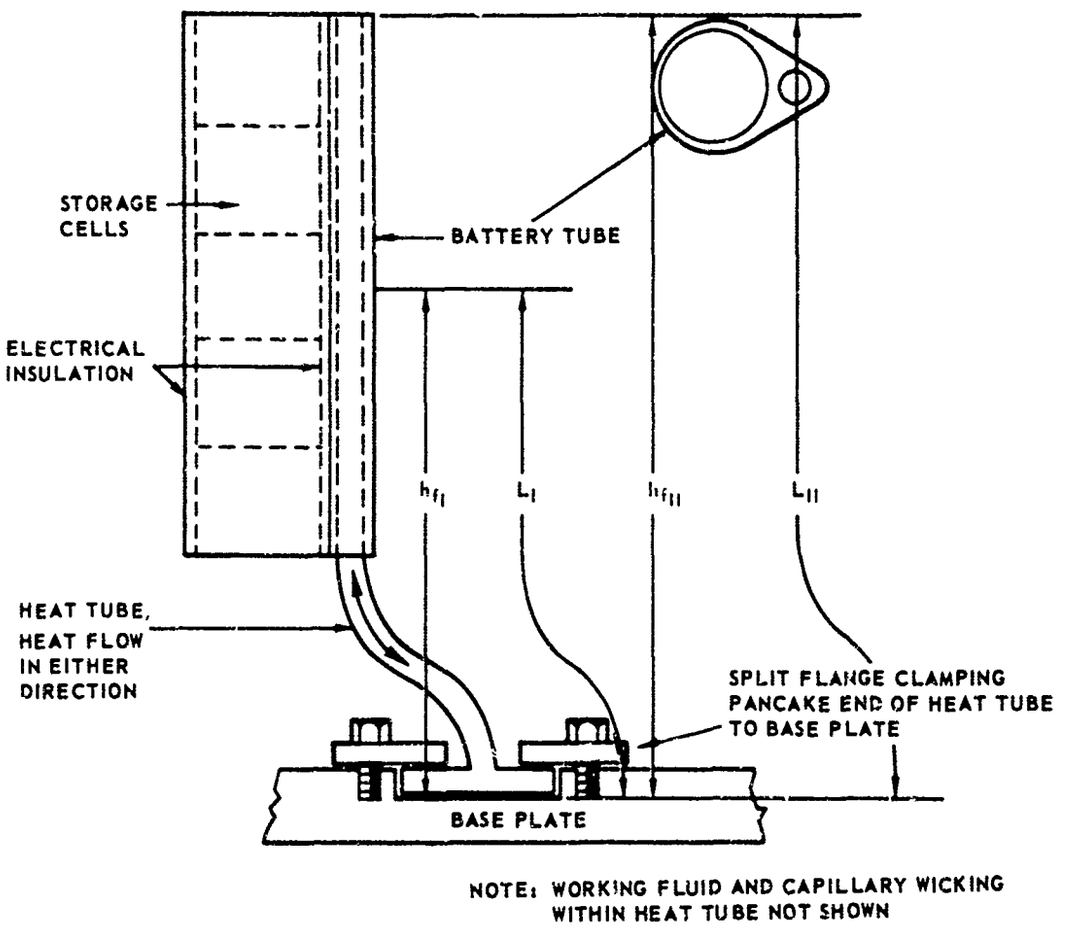

Fig. 3 SUGGESTED METHOD OF THERMALLY CONNECTING BATTERY TUBE TO BASE PLATE

The radical effect of the presence or absence of a gravitational or other force field is apparent. In performing these calculations it is noted that k is composed of $h_f \rho_\ell a \approx 2 \times 10^4$ and $(P_1 - P_2) = 3 \times 10^2$. Thus the effect of a modest temperature or pressure drop along the tube on the choice of D and on the value of $\dot{V}$ realized is slight in the gravitational case but must be allowed for conservatively in the orbital case.

One might design a heat pipe to operate optimally against 1 g so as to permit operation of all other portions of the satellite at the design point in ground testing. The Case I design employing the 1 g choice of D would, in the 0 g environment, be capable of $\dot{V} = 9.9 \times 10^{-2}$. Similarly, Case II would be capable of $\dot{V} = 1.78 \times 10^{-2}$ in orbit. This course of action would have the advantage of ground checkout in the most adverse attitude, but with a substantial overdesign for orbital use. The only apparent justification would be some stringent limitation of a ground test set-up.

A more desirable procedure in the case of an orbital application employing a straight pipe would be to design for 0 g and to check the operation of the heat pipe in a horizontal position so as to get effectively 0 g in the principal dimension.

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13. ABSTRACT

A heat pipe is a device which exhibits an extremely high effective thermal conductivity by means of two-phase fluid flow with capillary circulation.

The primary objective of the experimental program was to determine a suitable method of control for the heat pipe and to establish suitable wick/fluid configurations for the various temperature ranges of interest.

The primary objective of the prototype program was to provide design, construction, testing for verification, and flight hardware specifications of a heat pipe applicable to thermal control of a spacecraft or a spacecraft subsystem. Thus, a thermal design improvement for spacecraft could be proposed; in addition, thermal resistances of heat pipes could be derived.

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